ABSTRACT

Advances in technology and trends towards higher processing speeds have generated a greater need for thermal management. Two-phase cooling (boiling) has the ability to dissipate large amounts of heat and is attractive because of lower mass flow requirement and uniform substrate temperature. Further improvements can be obtained through passive surface enhancements. The objective of this work is to investigate the effect of microchannel surfaces on pool boiling performance at atmospheric pressure with FC-87. Being a dielectric fluid with a low normal boiling point, FC-87 has desirable characteristics for an electronics cooling fluid. A maximum heat flux of 550 kW/m² at a wall superheat of 37°C was obtained with the microchannel surface. Surface area increase is noted as the primary reason for the enhanced performance for FC-87 on microchannel surfaces.

1. INTRODUCTION

Direct immersion cooling is an attractive option due to its ability to remove large amounts of heat. Since the fluid would be in direct physical contact with the electronics it must be dielectric, stable, non-toxic, non-flammable and have a low boiling point. Different enhanced surfaces and working fluids have been explored by researchers for chip cooling application. FC-87 is a member of the Fluorient series manufactured by 3M, specifically designed for electronics cooling.

Passive enhancements have also been explored by various researchers for augmentation of pool boiling performance. Mudawar and Anderson [1] investigated various enhancement techniques on FC-72 and FC-87. The authors also studied the effects of system pressure, subcooling and surface area augmentations. For cylindrical enhanced surfaces, a maximum heat flux of over 100 W/cm² at a wall superheat 62.8°C was obtained for saturated FC-72. For FC-87, a maximum heat flux of 89.6 W/cm² was recorded at 50°C wall superheat. Chang and You [2] used aluminum brushable ceramic-M.E.K. (ABM) coating on tubular surfaces. The ABM coated surfaces showed performance enhancement with FC-87 and R-123. Large decrease in incipient wall superheat was also observed. Coursey et al. [3] used graphite foam with FC-72 and FC-87 at atmospheric pressure. Heat flux of around 145 W/cm² was obtained at a wall superheat of 52°C with FC-87. Klett and Trammell [4] also used graphite foams but in a slotted pattern to obtain a maximum heat flux of 150 W/ cm² at 11°C wall superheat. Graphite foam has a thermal conductivity of up to five times higher than that of copper, which constitutes significantly to the improved overall pool boiling performance. FC-72 comes from the same family as that of FC-87 (fluorocarbon based). Besides the difference in saturation temperature, the two fluids share similar thermophysical properties. FC-72 has been used as the working fluid by various researchers for pool boiling. Chang and You [5] studied the effect of coatings in saturated FC-72 at atmospheric pressure. They observed a significant decrease in incipience superheat with micro-porous surface. O’connor and You [6] used painted surfaces in saturated FC-72 to obtain low incipience superheats. Wei and Honda [7] used square micro-pin-fin with varying geometry in FC-72. A maximum heat flux of 84.5 W/cm² was recorded with a subcooling of 45°C. Guglielmini et al. [8] showed uniformly spaced fins in FC-72 performed better than plain surface. Longer fins with narrow fin widths and high system pressure showed better performance. Similar enhanced surfaces were also used by Yu and Lu [9] in their experimental investigation to study the flow patterns and pool boiling performance. Kim et al. [10] used a microporous coated surfaces in different fluids (R-123, FC-72 and water) at atmospheric pressure to experimentally study the effect of coating. Microporous surfaces showed significant improvement in pool boiling performance with FC-72 and R-123 over a plain surface.

ABSTRACT

Advances in technology and trends towards higher processing speeds have generated a greater need for thermal management. Two-phase cooling (boiling) has the ability to dissipate large amounts of heat and is attractive because of lower mass flow requirement and uniform substrate temperature. Further improvements can be obtained through passive surface enhancements. The objective of this work is to investigate the effect of microchannel surfaces on pool boiling performance at atmospheric pressure with FC-87. Being a dielectric fluid with a low normal boiling point, FC-87 has desirable characteristics for an electronics cooling fluid. A maximum heat flux of 550 kW/m² at a wall superheat of 37°C was obtained with the microchannel surface. Surface area increase is noted as the primary reason for the enhanced performance for FC-87 on microchannel surfaces.

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An in-depth literature review of pool boiling enhancement techniques with water as the working fluid can be found in Cooke and Kandlikar [11,12]. They showed that the microchannel surfaces are very effective in enhancing the pool boiling heat transfer with water. FC – 87 with its low normal boiling point, inert properties and dielectric nature can be considered as a viable option for electronics cooling. As seen from the literature, microchannels have performed extremely well in pool boiling with water. In the present work, microchannels are therefore further explored for pool boiling enhancement with FC – 87 as the working fluid at atmospheric pressure.

2. NOMENCLATURE

\( k_{Cu} \) \hspace{1cm} thermal conductivity of copper, W/m K
\( q'' \) \hspace{1cm} heat flux, W/m\(^2\)
\( x \) \hspace{1cm} distance, m
\( T_{sat} \) \hspace{1cm} saturation temperature, °C
\( T_c \) \hspace{1cm} chip temperature, °C
\( T_{wall} \) \hspace{1cm} wall temperature, °C
\( \Delta T_{sat} \) \hspace{1cm} wall superheat, °C
\( h \) \hspace{1cm} heat transfer coefficient, W/m\(^2\)°C

3. EXPERIMENTAL SETUP

The test setup used in this study was similar to the one used by Kalani and Kandlikar [13] in a previous study. The setup shown in Fig. 1 consisted of a stainless steel cylindrical chamber, 100 mm in diameter, which was sealed by two cylindrical flanges (top and bottom). Four C-clamps and O-rings on each side were used to ensure a leak proof system. A 120 VDC, 200 W auxiliary heater was used to maintain the FC-87 at the desired saturation temperature. An opening was also made on the bottom flange to allow the heater assembly to come in contact with the underside of the copper chip. Thermal conducting paste was used in between the contacting surfaces of the heater and the chip. A Garolite chip holder was designed and fabricated to hold the copper chip on the bottom flange by threaded bolts. Garolite was used due to its lower thermal conductivity (0.27 W/m K) and ability to withstand high temperatures (168°C).

Multiple openings for fluid inlet, thermocouple probe, vacuum port, pressure gauge and inlet and outlet connections for the copper condenser were provided on the top cylindrical flange. A pressure gauge was installed for pressure measurement inside the chamber. A coiled copper tube acted as a condenser for the current system. A constant temperature circulating water bath provided water flow into the condenser at the desired temperature and flow rate. For the current work, the vacuum port was not used and was sealed off.

The heater assembly is shown in Fig. 2. It consisted of a Garolite base, cartridge heater, copper rod and thermocouples. The Garolite base held the copper rod and the cartridge heater in place. A 120 V DC, 450 W capacity cartridge heater was fitted into the copper rod to provide heat flux to the test chips. Three K-type thermocouples spaced 8 mm apart on the copper block were used to measure the temperature gradient, which was used in calculating the heat flux.

The heat flux at the test section is calculated using 1-D conduction equation:

\[
q'' = -k_{Cu} \frac{dT}{dx}
\]

where the temperature gradient \( \frac{dT}{dx} \) is calculated using the three-point backward Taylor’s series approximation:

\[
\frac{dT}{dx} = \frac{3T_1 - 4T_2 + T_3}{2\Delta x}
\]

The actual heat flux to the test section was calculated by correcting the heat flux from Eq. (1) for the heat losses from the sides of the chip. The heat loss was less than 6.5 percent of the total heat flux. A National Instruments cDaq-9172 data

![Figure 1. Schematic of the pool boiling test setup.](image1)

![Figure 2. Schematic of the heater assembly.](image2)
acquisition system with NI-9213 temperature module was used to log the data. A LabVIEW® virtual instrument (VI) displayed and recorded the temperature and heat flux.

4. TEST SECTION

Microchannels were used as the enhanced surface as shown in Fig. 3 to study the pool boiling performance with FC – 87. The enhanced structure is similar to the one used in a previous study by Kalani and Kandlikar [13]. The chip overall dimension was 20 mm × 20 mm × 3 mm. A 10 mm × 10 mm × 2 mm groove was made on the underside of the chip to reduce the heat spreading effect. The groove promoted one-dimensional conduction heat transfer. The heat losses from the copper rod were reduced by wrapping it with a high temperature ceramic insulating sleeve. A 700 µm hole was made on the side of the chip for the insertion of the fourth K-type thermocouple, Tc. The temperature of the wall located at a distance of x1=1.5 mm from the thermocouple location is calculated using the heat flux obtained from equation (1):

\[ T_{wall} = T_c - q''\left(\frac{x_1}{k_{Cu}}\right) \]  

Figure 3.Microchanneled copper test section.

Microchannels of varying dimensions were machined on the top side of the chip using CNC machines. One plain and six microchannel chips of varying depth, channel width and fin width were selected to study the effect of microchannel geometry on the pool boiling performance. The dimensions were measured using a confocal laser scanning microscope and are given in Table 1. The average roughness value obtained for the plain chip was 0.14 µm. For microchannel, the roughness value at the top surface varies from 0.8 µm to 3.8 µm and at the bottom channel it was 0.6 µm to 3.2 µm. The variation in average roughness value was due to machining operation.

<table>
<thead>
<tr>
<th>Sr. No.</th>
<th>Type</th>
<th>Channel Depth (µm)</th>
<th>Channel Width (µm)</th>
<th>Fin Width (µm)</th>
</tr>
</thead>
<tbody>
<tr>
<td>-</td>
<td>Plain</td>
<td>0</td>
<td>N/A</td>
<td>N/A</td>
</tr>
<tr>
<td>1</td>
<td>Microchannel</td>
<td>456</td>
<td>207</td>
<td>193</td>
</tr>
<tr>
<td>2</td>
<td>Microchannel</td>
<td>470</td>
<td>194</td>
<td>402</td>
</tr>
</tbody>
</table>

5. UNCERTAINTY ANALYSIS

An uncertainty analysis was performed following the same procedure outlined in [12]. Temperature measurements were found to have the largest uncertainty. Uncertainty in calculating the surface temperature was 0.2°C. The largest uncertainty in the heat flux measurement was calculated to be between 6-10%.

6. EXPERIMENTAL PROCEDURE

The test setup was tested for leaks by evacuating the system to 10 kPa abs, pressure and monitoring it over a period of 24 hours for pressure changes. Less than 1 kPa pressure loss was noted that time period. The copper chip was fitted in place with the Garolite chip holder on the bottom flange. The copper heater was brought in contact with the copper chip with the help of a mechanical stand. Thermal paste was applied on the top surface of the copper heater to reduce the thermal contact resistance between the tip and the copper chip. Since the chip temperature was measured directly with the thermocouple Tc, contact resistance was not relevant in estimating the chip surface temperature or heat flux. The value of contact resistance was back calculated and found to be 8 X 10^6 m^2 K/W. However, this value is not used in any calculations as a thermocouple inserted in the chip was used to measure the chip temperature close to the boiling surface.

Liquid FC-87 was directly introduced through the fluid inlet opening. Vacuum port which was used previously for introduction of fluid is not used due to the low saturation temperature of FC-87. The liquid volume in the boiling chamber is adjusted to be at a level slightly above the auxiliary heater. The auxiliary heater and the copper heater were powered on to allow degassing of the fluid. During degassing the system pressure was maintained at atmospheric pressure with the help of the condenser. The inlet valve was opened partially for 10 seconds to allow the air to escape. The constant temperature water bath which was connected to the condenser was turned on and the temperature and flow rate were adjusted to maintain the desired system pressure.

Once the experiment was started, the voltage was increased in a step-wise fashion. At each step the readings of the five thermocouples were recorded. Steady state was determined when the thermocouples did not fluctuate more than ±0.1°C over a period of 10 minutes. The CHF was recognized by a change in the slope of the temperature rise curve. The tests were stopped and the immediate preceding value of heat flux during steady state operation was noted as the CHF value.

7. RESULTS

Experiments were performed on the seven copper chips listed in Table 1. For all tests, the pressure in the chamber was
kept at 1 atm. The results of seven copper chips; one plain and six microchannels tested in FC-87 at atmospheric pressure are presented in this section. Effects of the microchannel geometrical parameters and comparison with other enhanced surfaces from literature are also discussed. Repeatability of the experimental data was evaluated by running experiments under similar conditions at different periods. The results obtained showed difference within the range of the experimental uncertainties.

7.1 Plain and Microchannel Surface testing

The graph shown in Fig. 4 shows a boiling curve with heat flux on the y axis and wall superheat on the x axis. The heat flux is calculated using Eq. (1) where the projected area of the heater surface (100 mm$^2$) is considered for all tested chips. Wall superheat is defined as the difference in the surface temperature of the copper chip exposed to the working fluid and the saturation temperature of the working fluid. For the current work, wall temperature is calculated at the top surface of the microchannel.

Plain chip was tested first and a maximum heat flux of 350 kW/m$^2$ at 31°C wall superheat was obtained. Slight boiling overshoot was observed in the range of 15°C - 20°C wall superheat. Chips 1, 2 and 3 dissipated a maximum heat flux of around 550 kW/m$^2$ at a wall superheat of 37°C, while chips 5 and 6 recorded a maximum heat flux of around 510 kW/m$^2$ at 40°C wall superheat. Chip 4 showed the poorest performance compared to the other microchannel chips. Expectedly, plain chip performed the worst compared to all the microchannel chips. Very small boiling overshoot was observed for all microchannel chips. All tested chips followed a similar boiling curve pattern in which at lower heat fluxes, natural convection was the dominant mode of heat transfer. As the surface temperature increased and more nucleation cavities were activated causing more heat to be dissipated.

![Figure 4. Boiling curves of microchannel enhanced surface chips for FC - 87 at atmospheric pressure.](image)

Figure 5 shows the heat transfer coefficient plotted against the wall superheat of different chips at atmospheric pressure. Similar to Fig. 4, plain chip performance was the poorest compared to the other chips. Chip 1 obtained a maximum heat transfer coefficient of 17.5 kW/ m$^2$°C at 20°C wall superheat. The maximum heat transfer coefficient was observed in the 20°C – 30°C wall superheat range for all tested chips. This region is where fully developed nucleate boiling takes place. For wall superheat range between 30°C – 40°C, the heat transfer coefficient values remain constant and then taper off as the CHF is approached. In general, the pool boiling performance with FC-87 has been observed to be significantly lower than that with water. This is in agreement with results for other dielectric fluids reported in the literature. The main reason for the performance degradation is the lower thermal conductivity and lower latent heat of vaporization of the dielectric fluids as compared to water, which has best heat transfer characteristics.

The wall temperature, which represents the sum of the wall superheat and the saturation temperature at a given pressure, for FC-87, is well below the allowable threshold temperature limit for electronics. Chip 1 yielded a maximum heat flux of 550 kW/m$^2$ at a wall temperature of 67°C. Hence due to the lower saturation temperature of FC-87, although the overall values of wall superheat are much higher when compared with other fluids such as water and ethanol, the surface temperature are still within the acceptable limits for electronics cooling application.

![Figure 5. Heat transfer coefficients for the tested chips at atmospheric pressure.](image)

7.2 Effect of geometrical parameters

Table 2 shows the number of channels on each and microchannel chip and the corresponding surface augmentation factor. The surface area augmentation factor is the ratio of wetted area to the projected area of the heated surface (which is 100 mm$^2$). The wetted area depends on the channel depth, channel width and fin width for a given microchannel.

<table>
<thead>
<tr>
<th>Sr. No.</th>
<th>Type</th>
<th># of Channels</th>
<th>Surf. Area augmentation</th>
</tr>
</thead>
<tbody>
<tr>
<td>Chip 1</td>
<td>Plain chip</td>
<td>6</td>
<td></td>
</tr>
<tr>
<td>Chip 2</td>
<td>Microchannel chip</td>
<td>10</td>
<td></td>
</tr>
<tr>
<td>Chip 3</td>
<td>Microchannel chip</td>
<td>12</td>
<td></td>
</tr>
<tr>
<td>Chip 4</td>
<td>Microchannel chip</td>
<td>15</td>
<td></td>
</tr>
<tr>
<td>Chip 5</td>
<td>Microchannel chip</td>
<td>18</td>
<td></td>
</tr>
<tr>
<td>Chip 6</td>
<td>Microchannel chip</td>
<td>20</td>
<td></td>
</tr>
</tbody>
</table>
As seen from Table 2, Chip 1 has the maximum number of channels and the maximum surface area augmentation factor. As seen from Fig. 4 earlier, chips 1, 2 and 3 recorded a maximum heat flux of around 550 kW/m² at approximately 37°C. While chip 4 recorded the lowest heat flux compared to the other microchannel chips due to its low surface area augmentation factor. Chip 1, which has deeper channels, narrow pitch and the greatest surface area augmentation factor, yielded the best performance as seen from Fig. 5.

7.3 Effect of actual surface area
An insight into the mechanism responsible for heat transfer enhancement may be obtained by comparing the chip performance based on the actual heat transfer surface area. The effect of the surface area is also more clearly understood when the heat flux is normalized by the actual heat transfer area as seen in Fig. 6. The graph shows heat flux calculated using the actual area as opposed to the projected area seen in Fig. 5. It is seen from Fig. 6 that the boiling curves for all the microchannel chips fall below the plain chip curve. The surface area increase is thus seen as the dominant factor behind the improved performance of the microchannel chips for FC-87. Actual boiling heat transfer mechanism seems to be adversely affected for these microchannel chips.

![Figure 6. Boiling curve for the tested chips with heat flux normalized by the actual surface area.](image)

For water, Cooke and Kandlikar [12] have shown that the enhancement was mainly due to an improvement in the mechanism, and the surface area was not the major factor in the enhanced performance of the microchannel chips. Figure 7 shows the boiling curves obtained by Cooke and Kandlikar using actual surface area with one plain and ten microchannel chips with water at atmospheric pressure. They observed that the microchannel acted as liquid conduits and allowed water to rewet the surface at higher heat fluxes. This heat transfer mechanism was seen as the main reason for improved pool boiling performance with microchannels for water, while the enhancement in the case of FC-87 was mainly due to area increase, at a slight expense of the mechanism itself.

![Figure 7. Boiling curves from Cooke and Kandlikar [12] with heat flux normalized by the actual surface area.](image)

Similar observation of increase in surface area for improved boiling performance with FC-87 has been observed in literature. The CHF value of 89.8 W/cm² at 50°C wall superheat with FC-87 was recorded by Mudawar and Anderson [1] using a four pin surface. The authors were able to dissipate high heat fluxes due to the overall increase in the surface area compared to a plain surface.

Large wall superheats have also been observed in literature when testing with Fluorient series fluids. Chang and You [2] used micro-porous coated surface as their enhanced structure with FC-87 and R-123. The overall wall superheat decreased drastically. The authors used ABM coating on tubular surface to show the effect of micro-porous coating.

7.4 Finned Structure
A finned surface having a surface area augmentation factor of 12.5 was fabricated as seen from the Fig 8. The outer dimensions of the chip were the same as the microchannel ones, 20 mm × 20 mm × 3 mm. The dimensions of the fins were 8.2 mm in fin height, 1 mm in channel width and 0.4 mm in fin width.

![Figure 8. Finned Structure.](image)
Pool boiling test was conducted on this surface and the results were compared with the previously obtained microchannel surfaces. Fig 9 shows the heat flux vs. wall superheat for finned surfaces when compared with microchannel. A maximum heat flux of 1.25 MW/m$^2$ was recorded at 40°C wall superheat. The maximum heat dissipated by finned surface is approximately over 3.5X as that of plain surface and twice as that of the microchannel ones.

8. CONCLUSIONS

An experimental investigation of pool boiling performance of microchannel enhanced surface with FC-87 at atmospheric pressure was conducted. The following conclusions are drawn based on the study:

a) Microchannel chips dissipated 1.5X or more heat flux compared to a plain surface. A maximum heat flux of 550 kW/m$^2$ at a wall superheat of 37°C was obtained with chips 1, 2 and 3.

b) Slight temperature overshoot was observed for all tested chips.

c) FC-87 is a suitable candidate for electronics cooling due to its dielectric, low saturation temperature and inert nature. The maximum wall temperature with FC-87 at atmospheric pressure while dissipating 550 kW/m$^2$ was 67°C.

d) Surface area increase is the dominant factor in the improved boiling performance with microchannel surfaces for FC-87. Further increase in the overall surface area augmentation factor is expected to improve the performance.

e) Large wall superheat observed while using FC’s may be reduced by using micro-porous structure on the surface.

Surface area increment was identified as the dominant mode in obtaining high heat dissipation rate. Based on this, new finned surface with tall fins of 8.2 mm height was developed. The finned structure gave a maximum heat flux of 1.25 MW/m$^2$ at 40°C wall superheat and a heat transfer coefficient of 30 kW/m$^2$°C for the same wall superheat.

9. REFERENCES


