Enhanced Pool Boiling For Electronics Cooling Using Porous Fin Tops on Open Microchannels With FC-87

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ABSTRACT

The miniaturization trend in microelectronics demands effective heat removal from the high energy density components. Pool boiling in vapor chambers is a cooling technique which uses phase change to remove heat directly from the surface to effectively manage the thermal heat transfer needs. The enhancement in pool boiling can be achieved by providing additional nucleation sites and/or increasing the heat transfer surface area. In this paper, an enhanced copper surface with porous fin tops on open microchannels is used with FC-87 as the working fluid. A maximum Critical Heat Flux (CHF) of 37 W/cm^2 is achieved here which translates to a 270% enhancement in CHF compared to a plain chip. High speed images obtained suggest that separate liquid-vapor pathways generated from nucleation on the porous fin tops surface was responsible for the enhancement. The cohesive effect of channel width and channel depth is also studied here.

Keywords

Boiling enhancement; Microchannels; Porous; CHF; HTC

1. INTRODUCTION

The latent heat transfer involving phase change is an attractive cooling option used to cool many high energy density electronic components. Pool boiling is identified as an efficacious method to remove heat from these components but it is limited by the critical heat flux (CHF). CHF is a condition where the bubble coalescence leads to a vapor film that engulfs the surface and limits the heat transfer. A typical pool boiling curve relating the heat flux and the wall superheat is used to characterize the heat transfer performance. Heat transfer coefficient (HTC) is then deduced from the boiling curve to determine the heat removal capability of the surface. Increased CHF and HTC are desired for effective cooling performance.

Water with its excellent thermal properties coupled with its nontoxic nature is the most widely used liquid for boiling heat transfer tests. Water also serves as a baseline comparison for different surfaces developed by researchers. Recent developments have shown that boiling heat transfer performance is enhanced by incorporating microscale and nanoscale features on the heater surface. The resulting liquid wettability changes affects the nucleation characteristics and the heat transfer performance. Kruse et al. [1] fabricated an enhanced metallic surface using a femtosecond laser process technique. Hierarchical micro/nanostructures were incorporated on the heater surface. A CHF of 122 W/cm² was achieved, similar to a plain surface, however the HTC increased by 7.5 folds. A superwicking surface with area augmentation and increased number of nucleation sites contributed to the enhancement in this geometry. Pal and Joshi [2] conducted experiments at sub-atmospheric pressures to investigate the liquid fill-level on the thermal performance of a closed-loop dual chamber thermosyphon. A maximum heat flux of 111 W/cm² was

reported at a surface temperature of 83 °C. Cooke and Kandlikar [3] fabricated an open microchannel geometry on a copper surface to enhance the heat transfer performance. High speed images suggested that the channels contributed by improving liquid circulation in the region surrounding the departing bubbles. A maximum heat flux of 244 W/cm² was reported at a wall superheat of less than 10 °C. Similar area enhancement techniques have been reported in literature; all of which have resulted in significant enhancement in either CHF or HTC or both.

In addition to surface enhancement techniques, micro/nano porous coatings and nanofluids on the heater surface have proved to be effective by providing additional nucleation sites. You et al. [4] studied the effect of nanofluids on critical heat flux in pool boiling. Alumina nanoparticles dispersed in distilled and deionized water constituted the nanofluid. A 200% increase in CHF was reported and they concluded that the enhancement was not related to the increased thermal conductivity of nanofluids but instead a thin coating of the nanoparticles was observed to be deposited on the surface. Forrest et al. [5] used a layer by layer deposition technique to coat nanoparticles on a nickel wire. A 100% enhancement was obtained in both CHF and HTC. Surface wettability change without any change in the roughness was identified as the main reason for the improved performance. Their study also suggested that CHF increased with an increase in the thickness of the coating.

Concurrent to water, fluids like refrigerants, alcohols, binary mixtures and fluorinert series by 3M® have gathered keen attention of researchers due to their dielectric properties, and lower saturation temperatures. Mudawar and Anderson [6] conducted a series of tests to investigate the effect of system pressure, subcooling, surface augmentation and choice

of coolant on the pool boiling performance. Their results indicated that FC-87 reduced surface temperature drastically compared to FC-72. The paper further suggested that surface augmentation was effective in enhancing CHF, however temperature excursions were noticed. Chang and You [7] developed a porous surface with coating thickness of 50 μm and a fin surface with a pitch and height of approximately 1.39 mm and 0.88 mm respectively to study the effect of boiling performance with FC-87. A heat flux of 15 W/cm^2 for porous coating and 24 W/cm^2 for finned structure was reported.

Kalani and Kandlikar [8] tested open microchannel geometry with ethanol as the working fluid at sub-atmospheric pressures. Their results suggested that a temperature overshoot was observed for all investigated surfaces with a highest performance of 90 W/cm^2 at a surface temperature of 84 $^{\circ}\text{C}$ with ethanol as the working fluid. Although the performance with ethanol is very good in comparison with the refrigerants, its flammability may pose concerns in some applications. Rainey et al. [9] experimentally evaluated the effect of porous pin fins on the pool boiling performance. The effect of horizontal and vertical heater orientation was also investigated. Experiments were conducted using FC-72 over a subcooling range of 0 $^{\circ}\text{C}$ - 50 $^{\circ}\text{C}$ and pressure range of 30 kPa - 150 kPa. They concluded that subcooling significantly enhances CHF.

A novel heat transfer surface combining open microchannels and porous coatings was developed by Patil and Kandlikar [10,11]. A two-step electrodeposition process was used to create porous coatings on fin tops only by applying high current density for shorter periods and low current density for longer periods. Bubbles nucleated in the porous coating on fin tops and created liquid circulation in the microchannels as shown in Fig. 1. This was

identified as the chief contribution mechanism for CHF enhancement. A CHF of 325 W/cm² at a wall superheat of 7.3 °C was reported with water as the working fluid.

Figure 1. Liquid-vapor pathway proposed by Patil and Kandlikar [11]

As seen from literature, area enhanced surfaces combined with porous coatings can shift the pool boiling curve to the left and also delay CHF thereby offering a wider operating range. This paper focuses on using chips fabricated by Patil and Kandlikar [11] and evaluating their pool boiling performance with FC-87, which is a widely-accepted fluid in electronics cooling industry. The hypothesis of the current study is based on the superior rewetting pathways provided by microchannels and additional nucleation sites provided by porous coatings to cohesively work towards enhancing the boiling performance. The cohesive mechanism will lead to separate liquid-vapor pathways which will aid in delaying CHF by continually supplying liquid to the nucleation sites. This will help meet the electronics cooling requirement by dissipating high heat flux and also satisfying the maximum temperature limit (of around 85 °C) imposed for such applications.

2. Experimental Setup

A test setup designed and fabricated by Kalani and Kandlikar [8] was used to investigate the pool boiling performance with FC-87 (Fig 2). The tests were conducted at standard atmospheric pressure. The main components of the test setup included a stainless steel chamber, a heater assembly and a condenser unit. The stainless steel chamber was 100 mm in diameter and contained the working liquid. Two cylindrical flanges were used to seal the cylindrical chamber on either ends. O-rings were provided between the chamber and flanges to prevent leakage and to maintain the pressure in the chamber at the desired level. Openings were provided on the top flange for FC-87 inlet, saturation thermocouple probe, vacuum port and condenser inlet and outlet connections. The condenser was connected to a water circulating bath which had a temperature range of -30 °C to 150 °C and also a flow control valve to provide the necessary flow rate. A pressure gauge was used in conjunction with the condenser flow rate/temperature control to maintain the pressure inside the chamber at the desired level.

Figure 2. Pool boiling closed loop experimental setup designed and fabricated by Kalani and Kandlikar [8]

The bottom flange consisted of an auxiliary heater (120 VDC, 200 – W) which was grooved in and also an opening for the copper heater rod to contact the test chip. The test chip was housed in a garolite casing and bolted to the bottom flange. This ensured that the test surface was held in the horizontal orientation at all times during the experiments. A cartridge heater inserted into the copper heater rod was connected to the main power supply to supply heat to the test chips. A thermal Interface Material (TIM) was used between the copper heater rod and the test chip to minimize contact resistance. Three K-type thermocouples were inserted into the holes, drilled 8 mm apart, in the copper heater rod to measure the temperature gradient. A National Instruments cDaq-9172 data acquisition system with NI-9213 temperature module was used to record temperatures.

3. Test Section

The test section consisted of a 20 mm × 20 mm copper surface as shown in Fig. 3. Open microchannels were machined on a central 10 mm × 10 mm boiling surface. On the backside of the boiling surface a 2 mm × 2 mm slot is provided to minimize heat loss and facilitate 1-D conduction to the test chip. A 0.76 mm hole is provided in the center (1.5 mm below the boiling surface) to accommodate the thermocouple to estimate the surface temperature directly from the known heat flux. The schematic also shows the microchannel parameters used in this study which are the channel width, channel depth and fin width respectively. The fin tops consisted of porous coatings deposited and bonded using a two-step electrodeposition process as described by Patil and Kandlikar [10]. In the first step, a high current density which varied between 300 mA/cm² and 650 mA/cm² was applied for a short duration of 15 seconds. The current density depended on the surface area of each microchannel geometry. The next step involved application of lower current density for a

longer duration of time. The first and second step aided in deposition and substrate bonding, respectively.

Figure 3. Test section and parameters used in this study (not to scale)

The heat flux to the test section is computed using the Fourier's 1-D conduction equation

$$q'' = -k_{Cu} \frac{dT}{dx} \quad (1)$$

The temperature gradient dT/dx is calculated from the three thermocouple readings using Taylor backward-difference series approximation

$$\frac{dT}{dx} = \frac{3T_1 - 4T_2 + T_3}{2\Delta x} \quad (2)$$

Where, T_1 , T_2 and T_3 correspond to the temperature at the top, middle and bottom of the copper rod respectively.

The boiling surface temperature is estimated using Eq. (1) as follows.

$$T_{\text{wall}} = T_4 - q'' \left(\frac{x_1}{k_{\text{Cu}}} \right) \quad (3)$$

were, T_4 is the chip temperature measured directly and x_1 is the distance between T_4 and the boiling surface and is equal to 1.5 mm for all the surfaces investigated here. The temperature at the top of the fin is considered while reporting the wall superheat in this study.

Five chips with different microchannel dimensions were used to study the effects the geometrical parameters on the boiling performance with FC-87 at standard atmospheric pressure. The dimensions of the channel width, channel depth and fin width were measured using a laser confocal microscopy and tabulated in Table 1. The focus of the current work is to evaluate the pool boiling performance with FC-87 at atmospheric pressure. A plain chip was used to compare the performance improvement. The roughness (R_a) of the plain chip was measured using a scanning laser confocal microscope at five different locations on the boiling surface and averaged to be 1.6 μm .

Table 1. Test matrix with dimensions used in this study

Chip no:	Channel width (μm)	Channel depth (μm)	Fin width (μm)
1	300	400	200
2	400	200	200
3	400	400	200
4	762	400	200
5	762	200	200

4. Uncertainty analysis and heat loss

An uncertainty analysis was conducted similar to Kalani and Kandlikar [8]. Precision and bias errors were considered to calculate the total uncertainty in heat flux. Each thermocouple was individually calibrated using an Omega thermocouple calibrator.

Thermal conductivity variation due to temperature and least count of the instrument used to measure the distance between the spacing on the thermocouples were also considered in the calculating the uncertainty.

Cumulatively, the precision and bias errors can be calculated using the equation below,

$$U_y = \sqrt{B_y^2 + P_y^2} \quad (4)$$

Each individual error is calculated using the equation below,

$$U_p = \sqrt{\sum_{i=1}^n \left(\frac{\partial p}{\partial a} * u_{ai} \right)^2} \quad (5)$$

Using eq. (5) the uncertainty in heat flux is

$$\frac{U_{q''}}{q''} = \sqrt{\left[\left(\frac{U_k}{k} \right)^2 + \left(\frac{3U_{T_1} * k_{Cu}}{\Delta x * q''} \right)^2 + \left(\frac{4U_{T_2} * k_{Cu}}{\Delta x * q''} \right)^2 + \left(\frac{U_{T_3} * k_{Cu}}{\Delta x * q''} \right)^2 + \left(\frac{U_{\Delta x}}{\Delta x} \right)^2 \right]} \quad (6)$$

The total uncertainty in CHF, which is main region of interest, was estimated to be 9.5 %.

Fig. 4 shows a plot of heat flux versus wall superheat for chip 3 with error bars.

Figure 4. Uncertainty in heat flux in Chip 3

The design of the test chip and its assembly in the garolite casing demands area outside the boiling surface which results in heat loss to the ambient air due to natural convection.

A heat loss study was performed in which a fiberglass insulation is made to cover the boiling surface and a single phase test is run in air to calculate the heat loss. The heat loss is plotted against the difference in temperature between the surface and ambient air. A linear line fit equation is established and the corrected heat flux is computed. The heat loss did not exceed 2 W/cm^2 for all the tested surfaces here.

5. Experimental Procedure

The primary concern in any vapor chamber is to ensure complete sealing. To check for leakage, FC-87 was filled in the steel chamber and allowed to stand for a 24 hour period. After checking for the leaks, the chamber was evacuated and again filled with FC-87 by opening the inlet valve to a height of approximately 60 mm above the test chip. The test section and auxiliary heaters were powered by separate power supplies.

The voltage in the power supply was increased in small increments once saturation temperature was attained. The temperatures, coolant flow rate and pressure data were logged. The data was recorded after ascertaining steady state when the thermocouple readings did not fluctuate by more than $\pm 0.1 \text{ }^\circ\text{C}$ over a 10 min period. Standard atmospheric conditions were attained inside the chamber by regulating the coolant temperature and flow rate in the circulation bath, and power supplied to the heater.

6. Results

Pool boiling curves relating the heat flux and wall superheat were used to characterize the heat transfer performance of the tested surfaces. Wall superheat is the difference between the surface temperature of the test chip and the saturation temperature of the boiling fluid. Heat flux calculations were based on the projected area and the temperature at fin tops is used while reporting the wall superheat. Heat flux is represented in W/cm^2

since it is the most commonly employed units in electronics cooling applications. It is expected that the surface enhancements will shift the curve to the left compared to a plain chip.

Pool boiling tests were conducted on the test surfaces shown in table 1. The effect of channel width and channel depth of open microchannel configuration on the pool boiling performance is studied here. Firstly, a plain chip was tested for enhancement comparisons and it reached a CHF of 11 W/cm² at a wall superheat of 29 °C as shown in Fig. 5. The figure also shows the pool boiling data obtained for the chips 1-4. The fin widths were maintained constant, and the channel width and the channel depth are the only variable parameters in these tests.

Figure 5. Pool boiling curve obtained with FC-87 at standard atmospheric pressure.

In the enhanced chips, a CHF of 37 W/cm² was obtained at a wall superheat of 27 °C for chip 3 which had channel width of 400 μm, channel depth of 400 μm and fin width of 200 μm. This represented an enhancement of 270% in CHF compared to a plain chip. Chip 1 had a CHF of 31 W/cm² at a wall superheat of 25 °C. Chip 2, 4 and 5 had CHF's of 22

W/cm², 10 W/cm² and 19 W/cm² at wall superheats of 25 °C, 16 °C and 14 °C respectively. All the chips tested here showed enhancement in either CHF or HTC or both. This can be seen by the curves shifted to the left when compared to a plain chip in the pool boiling curve.

Figure 6. Heat transfer performance curve obtained for FC-87

Figure 6 shows the pool boiling HTC for the chips tested here. In this plot, HTC is plotted on the y-axis and heat flux on the x-axis. HTC is represented in SI units of kW/m²°C which is commonly used for performance enhancement comparisons. The effectiveness of the surface in terms of heat dissipation performance can be inferred from this plot. Chip 3 had the highest HTC of 20 kW/m²°C before reaching CHF. The general trend indicated that HTC increased with increasing heat flux till the inflection point which is seen to be between 15 - 20 W/cm² for most chips and then a dip in HTC values is observed beyond this point. All the tested surfaces had a higher HTC than plain chip suggesting that the porous fin tops on open microchannels significantly improved the performance.

6.1 Effect of channel width

Two geometric parameters of open microchannels are studied here. The effect of channel width can be understood from the performance of chips 1, 3 and 4 as shown in Fig. 7. In this plot, CHF is plotted on the y-axis and channel width on the x-axis to find a trend in the performance characteristics. The CHFs for chips 1, 3 and 4 indicate that there is existence of an optimal channel width for a given channel depth and fin width. Chip 1 and chip 3 indicate that CHF increases with increase in channel width but drops significantly for chip 4 with a wider channel.

The variation of CHF with channel width for chip 2 and chip 5 with a constant channel depth = 200 μm and fin width = 200 μm is also studied. There is no stark difference in performance of these two curves but CHF of chip 2 is slightly higher than chip 5 which fits into the trend explained previously.

Figure 7. Effect of channel width and CHF for chips 1, 3 and 4

6.2 Effect of channel depth

Two groups can be formed to understand the effect of channel depth. Group 1 consisted of chip 2 and 3 which had a constant channel width of 400 μm and fin width of 200 μm , and group 2 consisted of chip 4 and 5 and had a constant channel width of 762 μm and fin

width of 200 μm . Group 1 suggested that the deeper channels (chip 3) performed better than the shallower channel (chip 2). Group 2 suggested the opposite was true where an increase in channel depth causes the CHF to drop significantly. This suggested that an optimum channel width to depth ratio exists for an open microchannel surface. This seems to be dependent on the location of nucleation sites, the liquid feed mechanism to the nucleation site and the properties of the working liquid.

Figure 8 shows a high speed image obtained at a heat flux of 5 W/cm^2 obtained with a photron fastcam® high speed camera at a frame rate of 4000 fps for chip 4. Similar images were obtained for chip 2 as well. The image identifies bubble nucleation on the fin tops and their subsequent rise. The liquid replenishment occurs from the liquid flow into the channels regions. This convective mechanism is similar to a jet impingement generating separate liquid and vapor pathways as shown in the schematic. At higher heat fluxes the nucleating bubble activity is intense due to availability of additional nucleation sites inside the porous coatings. This intense bubble nucleation activity enforces the liquid circulation and further reinforces the convective mechanism which is critical to enhance the performance. However, the vigorous nucleation and bubble departure activity inhibits visual access to the heater surface at higher heat fluxes. The heat transfer performance obtained is significantly higher with a CHF as high as 37 W/cm^2 and a HTC of 20 $\text{kW}/\text{m}^2\text{C}$ under this mode.

This mechanism reveals that the liquid transport in the channel region governs the CHF limit for all open microchannels with porous fin tops. The capability of liquid to overcome the flow resistance offered by the micorchannel walls is critical to delay CHF. There are many variables that affect the overall performance of the surfaces investigated here.

However the experiments conducted here suggests that the liquid flow resistance and liquid impingement effect are the two competing mechanisms. It is seen that the convective mechanisms are similar for both water [11] and FC-87 as shown here.

Figure 8. High-speed image of nucleation on the fin top surface (top image) and a schematic representation (bottom image) of liquid and vapor pathways for chip 4

Further examination of the results indicated that the channel width to depth ratio is an important parameter. Chips 1 and 3 have this ratio as 1 or lesser, while it is 2 for Chip 2, 1.9 for Chip 4, and 3.8 for Chip 5. For a ratio of 1, the liquid is seen to be able to efficiently irrigate the microchannel walls. If this ratio exceeds 1, liquid may not be able to reach the bottom of the microchannels and may lead to performance deterioration. This needs to be checked in further refining the fin geometry. However, as the width/depth ratio goes above

1, liquid circulation and reduction in area enhancement start to become a concern. The performance further deteriorates with the increased flow resistance which is encountered by the liquid in the channels when the ratio is above 1. Chip 5 with the highest width/depth ratio of 3.8 yields the lowest CHF in the group of enhanced chips. While the porous coatings provide a bubble pumping mechanism, liquid circulation and area enhancement in different microchannel configurations are important considerations in enhancing heat transfer performance. This paper addresses the liquid transport concerns in an open microchannel configuration by identifying a suitable width to depth ratio of the channels.

6.3 Comparison with water

The test results obtained with FC-87 are compared with water [11]. Table 2 summarizes the test results with channel width to depth ratio being the governing factor linked to the performance. A definite trend is not observed which can collaborate the performance of FC-87 and water. However it can be seen that when the channel width/depth ratio is 1, highest performance is achieved with water as well as FC-87. For FC-87, a channel width/depth ratio of less than 1 enhances the performance. A 3.3 and 2.8 fold CHF enhancement is achieved for a ratio of 1 and 0.75 respectively. Reducing this ratio below 1 increases the available surface area, but liquid circulation is adversely affected.

Table 2. Comparison of CHF enhancement for FC-87 and water

Channel width/channel depth	CHF _{enhanced} /CHF _{plain}	
	FC-87	Water
0.75	2.81	2.2
1	3.36	2.56
1.9	0.9	2.2
1.72	1.72	2.1
2	2	1.89

6.4 Comparison to literature

Figure 9 shows the pool curves for some of the top performing surfaces available in literature and chip 3 used in this study with FC-87 as the working liquid. Two surface enhancements, porous coatings and fins, were developed by Chang and You [7]. These structures had a CHF of 14 W/cm² and 24 W/cm² respectively. Mudawar and Anderson [6] developed fins that were approximately 12 mm tall and reported a CHF of 21 W/cm² with FC-87. Enhancement in this configuration is mainly driven by area augmentation. However, the best performing chip (Chip 3) in this study had a CHF of 37 W/cm² which was the highest reported CHF in literature with FC-87 at atmospheric pressure. The enhancement mechanisms can be attributed to the additional nucleation sites, liquid circulation within the microchannels, and the additional heat transfer area available over the microchannel surfaces.

Figure 9. Pool boiling performance comparison with available curves in literature [6, 7]

Patil and Kandlikar [11] proposed a mechanism in which the bubbles generated on the fin tops induced liquid circulation within the microchannels with water. This

mechanism is seen to hold good for FC-87 as well based on the image shown in Fig. 8 and the performance obtained. The strategic location of nucleation sites on the fin tops seems to increase the vapor generation rate with liquid supply through the channel regions of microchannels. The combination of two techniques enhances the performance by facilitating separate liquid-vapor pathway which is identified as the key mechanism to delay CHF and increase HTC.

8. CONCLUSIONS

A combination of two enhancement techniques was employed on copper chips to study their pool boiling performance with FC-87. Five open microchannel configurations with porous fin tops are tested at standard atmospheric pressure in a closed loop vapor chamber. The effect of channel width and depth were also investigated. The following conclusions are drawn from this study:

1. Chip 3 with a channel width and depth of 400 μm had the best performance amongst all chips investigated here with a CHF of 37 W/cm^2 . This represented a 270 % enhancement in CHF compared to a plain chip. A maximum HTC of 20 $\text{kW}/\text{m}^2\text{C}$ was obtained for the same chip.
2. The effect of channel width on CHF was studied. CHF increased with increasing channel width but dropped significantly as the width increased beyond a certain value. As the width increased from 300 μm to 400 μm , the CHF increased, but reduced dramatically for a width of 762 μm . Increasing channel width provides better liquid circulation, but reduces the heat transfer surface area. The results suggest that an optimum channel width exists for a given channel depth and fin width.

3. The effect of channel depth was also investigated here. For a channel width of 400 μm , deeper channels performed better than shallower channels. When a channel width of 762 μm is considered, shallow channels performed better. This suggested that the deterioration or enhancement in performance is directly linked to the channel depth and width where the rewetting pathways in open microchannels is heavily influenced by the competing mechanisms offered by the microchannel geometry.
4. The ratio of channel width to depth is identified to be an important parameter for performance improvement. Having a ratio less than 1 assures good liquid circulation along with area enhancement, while increasing it further has a detrimental effect due to a reduction in the heat transfer surface area and increase in flow resistance.
5. High speed images obtained at 4000 fps show bubble nucleation on fin tops while no nucleation was observed in the channel region. This behavior was similar to that observed for water in [11] suggesting that bubble nucleation occurred on the porous fin tops with subsequent liquid addition through a jet impingement like mechanism in the channel regions. A channel width to depth ratio of 1 provided the highest performance among the chips tested for both water and FC-87.
6. Additional nucleation sites and bubble pumping mechanism provided by porous coatings and separate liquid-vapor pathways provided by open microchannels work cohesively to improve the pool boiling performance.

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NOMENCLATURE

k_{Cu}	thermal conductivity of copper, W/m°C
q''	heat flux, W/m ²
T_{sat}	saturation temperature, °C
T_{wall}	wall temperature, °C
ΔT_{sat}	wall superheat, °C
x	distance, m

Figure Captions List (w/ fitting preference)

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