

Multi-nozzle Closed Loop Spray Cooling Systems in Electronics Cooling

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Abstract. This study presents two refrigerant closed loop spray cooling systems for two specific electronic cooling applications. A small system using a 3×2 nozzle array is developed to investigate the spray cooling performance on a concentrated heat source ($2 \times 1 \text{ cm}^2$) with a high heat flux. A big system uses a 9×6 nozzle array to cool a 6U card area ($23.3 \times 16 \text{ cm}^2$) with a moderate uniform heat flux. The experimented flow rate in the small system is maintained at $7.8 \sim 8.1 \text{ g/s}$ corresponding to a spray mass flux of $3.9 \sim 4.05 \text{ g/cm}^2 \cdot \text{s}$ on the concentrated heat source, and the studied flow rates in the big system are monitored at $113 \pm 4 \text{ g/s}$ corresponding to a spray mass flux of $0.30 \text{ g/cm}^2 \cdot \text{s}$ on the 6U card surface area. The results show that the small system with a much higher spray mass flux can perform a spray cooling curve with a wide heat flux range. The big system with much less spray mass flux has performed a slightly better heat transfer coefficient due to its much better evaporation efficiency at the same degree of surface superheat.

1. Introduction

In our current technological age, there is a vast increase in the usage of electronic equipment on a daily basis, ranging from large scale circuit boards and small scale micro-processors. A majority of these equipment or devices generate waste heat during operation, and this excess heat has to be removed timely in order for the equipment or devices to sustain their optimum operating temperature. Many thermal management schemes have been proposed for removal of high heat fluxes from power electronic devices, such as thermosiphons, channel flow boiling, jet impingement, and spray cooling. Among them, spray cooling is gaining more popularity due to its prominent features such as high flux heat dissipation, uniform surface temperature, and ability to cool large surface area using a single nozzle or multiple nozzles array [1].

In literature, most of the experimental studies in spray cooling have been associated with free space spray to cover a small cooling area [2, 3] using a single spray nozzle. However, only a few studies have used multiple nozzles to cool large surface area in closed loop systems [4-6]. Using a multi-nozzle array to cool a large surface area encounters the liquid management challenges in both of supplying liquid to spray nozzles and draining excessive liquid on the heated surface. These challenges will be more crucial when a larger flow rate is demanded to reject higher heat flux in a closed loop. Moreover, a closed loop system embodies more equipment working under high pressure, which makes the system very hard to control and somehow limit the applicability of spray cooling technology in continuous operations. So far, pioneer research has been conducted to build closed loop spray cooling systems for different applications. Lin et al. [4] developed a spray cooling system by



using a magnetic gear pump to circulate a closed loop. In their system, the heated surface was impinged by the atomization from an array of 48 jet-swirl nozzles in a compact spray chamber. FC-72 was the working fluid. Their experiments showed that the spray cooling performance was affected by several factors such as flow rate, uprising vapour from the heated surface, and unsteady inlet flow. The liquid flooding in the spray chamber impairs the heat transfer process and prevents the system from removing heat effectively. Yan et al. [7] developed a closed loop spray cooling system with four gas-assisted nozzles to simulate the spray cooling on an electronic card. The flow loop was driven by a gas compressor and R-134a was the working medium. It was reported that this system was able to remove 1 kW from the heated surface without exceeding the mean surface temperature of 25°C. Tan et al. [8] developed a closed loop spray cooling system using a refrigerant cycle. An array of 3×2 nozzle plate was applied to remove heat from a concentrated heat source up to 150 W/cm². They reported that the heat source temperature can be maintained below 50 °C at a heat flux up to 150 W/cm², which shows promising potential for thermal management of temperature sensitive power electronics.

In the present study, two closed loop spray cooling systems were developed to cool a small surface area with a heat flux up to 160 W/m², and a 6U card area with a heat load up to 16 kW, respectively. The heat transfer performances of these two closed loop systems are presented and discussed. This study will shed some light on developing closed loop spray cooling systems for different applications in industry.

2. Experimental Setup

The proposed closed loop spray systems in the present study are modified from two normal refrigerant cycles with different cooling capacities (1 kW and 52 kW, respectively). These two systems are designed based on the same principle which is illustrated in Fig.1. The system mainly comprises of an inverter compressor, a condenser, a liquid-vapour separator, a spray chamber, an accumulator, and other accessories. In operating the system, the saturated or sub-cooled liquid refrigerant exiting from the condenser goes through the metering valve to form saturated vapour and liquid in the liquid-vapour separator. The separator separates the liquid to the bottom and the separated liquid is then channelled to the feeding chamber to supply the liquid to the spray nozzles uniformly. The droplets atomized at the nozzles impinge on the heater surface to absorb heat, and the vaporized refrigerant as well as the excess liquid then flows back to the gas-liquid accumulator where the refrigerant is sucked back to the compressor in the form of vapour for the next cooling cycle.

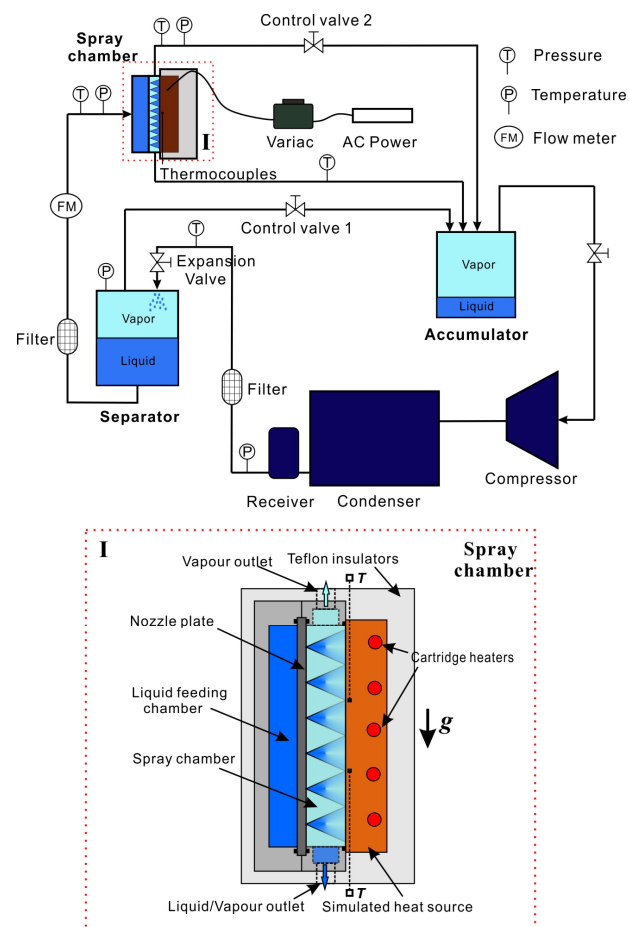


Fig. 1. Schematic of closed loop refrigerant spray cooling systems.

The critical part in such closed loop systems is the spray chamber which is carefully designed to establish proper conditions for multi-nozzle array spray cooling to take place over a heated surface. As shown in Fig.1, the spray chamber is assembled vertically with a liquid feeding chamber, a nozzle plate, a spray chamber, a simulated heat source and Teflon insulators. Hence, gravity is applied to assist the liquid drainage from the heated surface. Specifically, in the small system (1 kW), the heated surface has an area of $2.0 \times 1.0 \text{ cm}^2$ which is covered by a 3×2 nozzle array as shown in Fig.2(a). The nozzle plate features a housing plate and jet-swirl inserts to form the jet-swirl nozzle array. In this system, the nozzle-to-surface distance was fixed at 0.88 cm which makes the spray cones just inscribe the heated surface and a very compact spray chamber. In the big system (52 kW), the simulated heat source has a rectangular impinged surface area (cooling area) of $23.3 \times 16.0 \text{ cm}^2$ which is the exact size of a 6U electronic card. The nozzle plate integrates 54 commercial pressure swirl nozzles that have been configured with an array of 9×6 . The nozzle-to-surface distance is fixed at 2.3 cm which makes the atomization of the nozzle array be fully developed to cover the 6U card area. In both systems, the simulated heat sources are designed according to ANSYS steady-state thermal simulations. It ensures a reasonable assumption that the heat flux over the heated surface is uniform and can be correctly determined by the overall heat load applied.

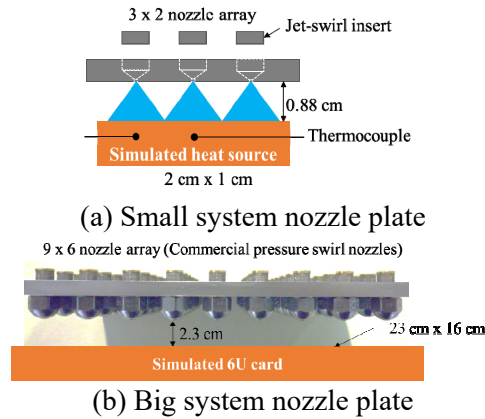


Fig. 2. Configurations of nozzle plates in the big and small systems.

In experiments the mass flow rate, pressure difference across the nozzles plate and chamber pressure are controlled by regulating the compressor frequency, metering valve and the control valves 1 and 2 in Fig.1. The total mass flow rate across the nozzles plate was measured according to the digital flow meter in the upstream of the spray chamber. The applied heat load on the heat sources are controlled by regulating the joule heat generated by the installed cartridge heaters. Four thermocouples are installed in the small system to measure the averaged surface temperature of the 2 cm² surface, and 27 thermocouples are installed in the big system to calculate the averaged surface temperature on the 6U card surface.

3. Data reduction

In the present study, the heat load applied to the heated surface is calculated by

$$Q = V \cdot I \quad (1)$$

where V is the voltage and I is the current in the electric circuit of cartridge heaters. Therefore, the heat flux (q) applied on the heated surface is calculated as

$$q = Q/A \quad (2)$$

where A indicates the heated surface area. According to the assumption of isotherms below the heated surface, the local surface temperatures can be determined as

$$T_{surf,i} = T_{thermo,i} - q \cdot \Delta x / k \quad (3)$$

where $T_{thermo,i}$ is the local temperature reading of the i th thermocouple, k is the thermal conductivity of the heater block, and Δx is the distance (0.4 cm in this study) from the thermocouple plane to the heated surface. The mean surface temperature is thereby calculated as the arithmetic averaged value of the local surface temperatures,

$$T_{surf} = (T_{surf,i} + T_{surf,i+1} + \dots + T_{surf,N})/N \quad (4)$$

where N is the number of measurement points. The average heat transfer coefficient h is thereby calculated as

$$h = q / (T_{surf} - T_{sat}) \quad (5)$$

In order to study the liquid usage efficiency in the experiments, evaporation fraction (ε) which represents the proportion of liquid evaporated in the spray and heat transfer process with respect to the total liquid flow rate is evaluated. Considering the spray chamber as a control volume, the evaporation fraction is defined according to the energy and mass conservations in the spray chamber, and the equations can be found in reference [5].

4. Results and Discussion

In this study, the investigated flow rate in the small system is maintained at 7.8~8.1 g/s which is corresponding to a spray mass flux of 3.9~4.05 g/cm²·s on the small heated surface. In the big system, the investigated flow rates is kept around 109~117 g/s which is corresponding to a spray mass flux of 0.29 to 0.31 g/cm²·s on the 6U card surface area.

As shown in Fig.1, the spray cooling curves of the big and small systems are illustrated. The relatively short spray cooling curve of the big system is due to the limited spray mass flux that can be supplied by the nozzle plate. As a result, the “dryout” occurs when the surface heat flux and temperature are moderate ($q = 42 \text{ W/cm}^2$ at $T_{\text{surf}} = 25.6^\circ\text{C}$). In the small system, as the spray mass flux is more than 10 times as compared to the big system, its spray cooling curve is remarkably extended. This indicates that the spray mass flux has significant effect on delaying the occurrence of critical heat flux. A higher spray mass flux can continuously provide sufficient droplets onto the heated surface to maintain a stable heat transfer in a much wider heat flux range. However, it is surprising that the spray cooling curves of these two systems are almost overlapped although their spray mass flux are more than one order of magnitude difference. Furthermore, it is found that the heat transfer coefficient of the big system is even slightly higher than that of the smaller system as indicated in Fig.4. This could be attributed to the better evaporation of the sprayed refrigerant in the big system as shown in Fig.5.

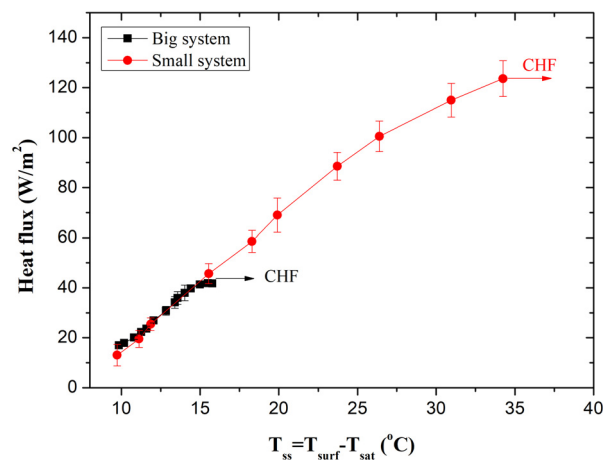


Fig. 3 Spray cooling heat transfer curves for both systems

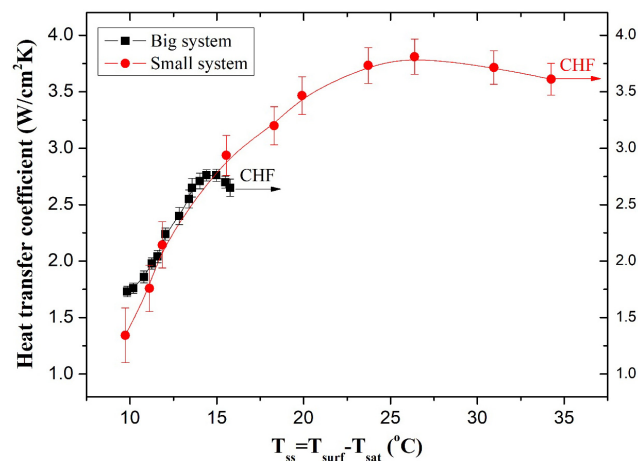


Fig.4 Heat transfer coefficient curves for both systems

The liquid evaporation efficiencies achieved in both systems are illustrated in Fig.5. It demonstrates that the big system having smaller spray mass flux on the 6U card heated surface can use the liquid refrigerant efficiently. Prior to the occurrence of critical heat flux, the liquid evaporation efficiency can reach as high as 0.87, which shows practically full evaporation of the available refrigerant. When the evaporation efficiency exceeds 0.87, CHF occurs as the remaining liquid refrigerant is not able to maintain a continuous liquid film on the heated surface. However, in the small system, it is clear that the sprayed liquid refrigerant is not efficiently utilized as the evaporation efficiency is only close to 0.32 before CHF happens. At the early stage of the spray cooling curve, the dense spray on the heated surface suppresses the evaporation of the sprayed refrigerant on the heated

surface (evaporation efficiency is less than 0.25). Hence, the heat transfer coefficient is poorer in comparison to the big system. As the heat flux increases, CHF is triggered by the drastic bubble boiling occurring on the heated surface where large bubble acts as an insulation layer to prevent the impinging droplets from impacting on the heated surface [8].

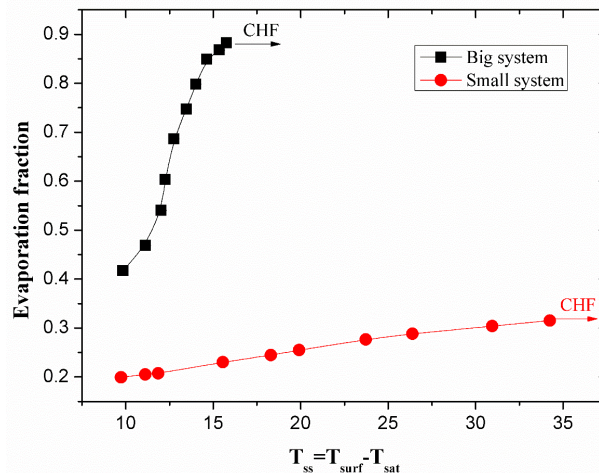


Fig.5 Evaporation efficiencies in the big and small systems

5. Conclusion

Two closed loop systems with different scales are built to study the spray cooling heat transfer performance in two specific applications. The small system applying a nozzle array of 3 x 2 is developed to test the refrigerant spray cooling on a concentrated heat source in a compact spray chamber. The big system applies 54 commercial pressure swirl nozzles to cool a 6U card with uniform heat fluxes. The results show that the small system with a much larger spray mass flux obtains a spray cooling curve with a wider heat flux range. However, the big system with much less spray mass flux has achieved slightly better heat transfer coefficients due to its better evaporation efficiency at the same degree of surface superheat.

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