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MULTI-NOZZLE SPRAY COOLING IN A CLOSED LOOP (POSTPRINT)

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ABSTRACT

A closed two-phase loop system was developed that combined with a multi-nozzle spray cooling unit for the cooling of high heat flux power sources. The fluid circulation was sustained by a magnetic gear pump operating with an ejector pump unit. The motive flow of the ejector shared the pumping liquid flow with the multi-nozzle spray. The use of the ejector stabilized the circulation of the two-phase flow. A multi-nozzle plate with 48 miniature nozzles was designed to generate an array of 4×12 sprays. A closed loop spray cooling experimental setup with a cooling area of 19.3 cm^2 was built. The spray nozzle to target distance was 10 mm. Water and FC-72 were used as the working fluids. Spray cooling experiments were performed in three orientations of the spray target surface, namely (a) horizontal facing upward, (b) vertical, and (c) horizontal facing downward. The thermal performance of the horizontal facing downward surface was the best. A comparison with the thermal performance data for a smaller cooling surface area of 2.0 cm^2 was made.

KEY WORDS

Spray cooling, two-phase heat transfer, closed loop, CHF

INTRODUCTION

The High Power Electronics are dependent on how well the large amount of waste heat can be managed in the system and effectively removed from the system. Direct cooling by means of jets and sprays was considered as a solution to the problem of cooling high power density direct energy devices since both methods were capable of removing high heat flux. Spray cooling had several advantages but its closed two-phase loop system for the aerospace application would be more complicated. In view of saving pumping power per unit of power removed, the microjet arrays proved superior to the sprays (Fabbri et al., 2005). From the point of view of keeping a low surface superheat and low coolant flow rate, spray cooling with phase change was exhibited to be a more effective method of removing high heat fluxes (e.g., greater than 500 W/cm^2 using water as working fluid) from surfaces.

A major portion of the spray cooling heat transfer results from nucleate boiling heat transfer within the thin liquid film produced by impinging droplets on the cooling surface. The accompanying heat transfer modes are convection and direct evaporation from the surface of liquid film. It was believed that the collapse of the vapor bubble in the liquid film either by the impingement of liquid droplets or by the merging of the bubble on the top of the thin liquid film accounted for the heat transfer enhancement of the spray cooling. In the past 20 years, many spray cooling experiments were performed to understand heat transfer characteristics and critical heat flux (CHF) at cooling surfaces (Sehmbey et al., 1992, Mudawar and Estes, 1996, Rini et al., 2002, Lin and Ponnappan, 2003, Horacek et al., 2004). Only a few results were reported in literature dealing with multi-nozzle spray cooling to remove high heat flux from a larger surface area at a level of 20 cm^2 (Lin et al., 2004) or higher. The thermal performance of the spray cooling over the large hot surface was different from the case of small area spray cooling. The thermal performance of the large area spray cooling was lowered because of a stronger interaction between the spray cones in the region adjacent to the cooling surface and an increased liquid film thickness on the cooling surface. Therefore, on the heat acquisition side of a large area spray cooling system, emphasis was placed not only on the multi-nozzle array configuration but also on the effective discharge of the two-phase fluid from the spray chamber. The vapor flowing towards the pump could result from insufficient subcooling of the fluid from the condenser, an insufficient fluid fill amount and unsteady two-phase flow.

EXPERIMENTAL SETUP & PROCEDURE

The experimental setup of multi-nozzle spray cooling in a closed loop is shown in Figure 1. The closed loop system mainly consisted of a preheater, a spray chamber housing a 48-nozzle assembly (in a 4 by 12 array), a heater assembly with a cooling surface area of 2.54 cm × 7.60 cm, a coaxial coil condenser, an ejector unit with the motive flow from the bypass loop, a magnetic gear pump, a liquid reservoir for liquid charge, and a filter. A cold bath was used to supply the cooling water to and from the condenser. The inlet port of the magnetic gear pump was connected with an extension of the discharge port of the ejector. FC-72 and water were used as the working fluid. The closed loop system was evacuated before filling the working fluid. The two-phase fluid from the spray chamber flowed into the condenser where the vapor condensed. The motive flow of the ejector drove the subcooled liquid from the condenser and they mixed in the ejector. The merging liquid flow was then pressurized through the magnetic gear pump. The liquid from the pump was divided into the flow for the spray cooling and the flow as the motive liquid for the ejector. The spray chamber (with the cooling surface) could be rotated in three orientations: (a) horizontal facing upward, (b) vertical, and (c) horizontal facing downward as shown in Figure 2. A cartridge heater was employed in the case of FC-72 while a Vortek plasma heater was used in the case of water. When the Vortek heater was used, the spray cooling surface was vertically placed. The spray distance was 10 mm. In the spray chamber, there was a side channel surrounding the spray nozzles to effectively remove the two-phase fluid away from the cooling surface. The liquid flow rates in the spray loop and the bypass line were measured using two turbine flow meters. All pressures were measured using pressure transducers. The spray pressure drop, $\Delta p = p_1 - p_5$, was controlled by the power input to the pump and the bypass valve. The spray chamber pressure, p_5, corresponded to the spray saturation temperature, T_{sat}. The fluid temperatures in the spray cooling system were measured using type T probe thermocouples. The liquid temperature at the inlet of spray chamber was regulated by adjusting the cold bath temperature and input power to the preheater. The ejector performance was characterized with three pressure transducers at each port of the ejector and two flow meters in the spray loop and the bypass line. During the experiment, the pressure drop across the spray nozzles was varied from 1.03 bar to 2.28 bar for water and to 3.80 bar for FC-72. The heat flux was calculated using the average temperature difference between the two thermocouple location planes and the thermocouple pair distance (along the heat flux direction) in the heater plate. The average temperature on the cooling surface, T_w, was estimated through the extrapolation of the thermocouple readings across the two planes.

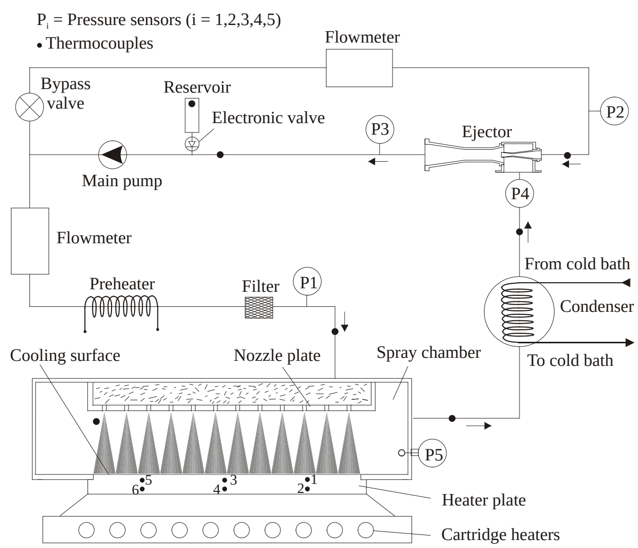

Figure 1. Schematic of the experimental setup.

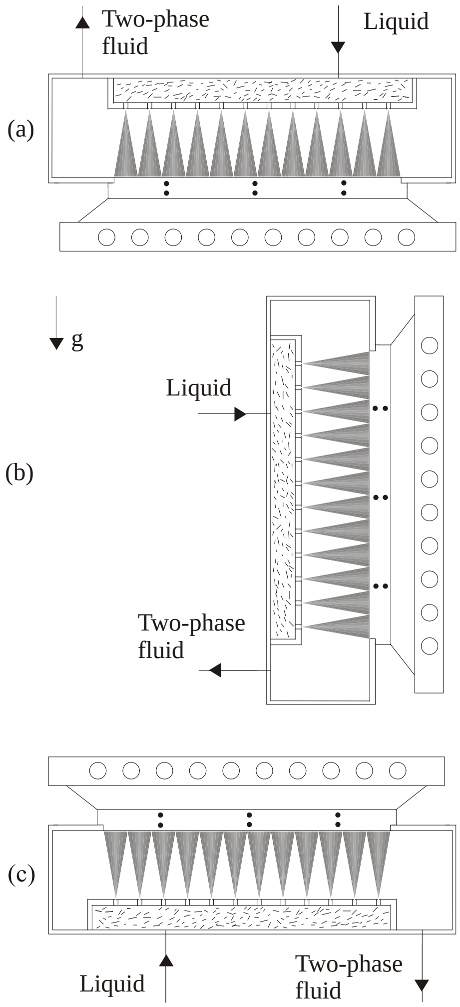

Figure 2. Spray cooling surface orientations:
(a) Horizontal facing upward, (b) vertical and
(c) Horizontal facing downward.

The data acquisition unit and the thermocouples were calibrated and the system accuracy was found to be within 0.2°C over the range of interest. The measurement accuracy of the distances between two thermocouples in each pair and between the cooling surface and its closer thermocouple locations in the heater plate was 0.2 mm. The uncertainty of the heat flux was 12% at $q''=10$ W/cm^2 which was the smallest heat flux applied. The uncertainty of the hot surface temperature, T_w, was estimated within 3.1°C at $q''=500$ W/cm^2 which was the upper limit in the present experiment. It was noted that the effect of the temperature gradient across the thermocouple beads in the heater plate was not considered in the uncertainty analysis. The accuracies of the pressure transducers reading were 8.6×10^{-3} bar. The spray saturation temperature was calculated as function of the spray chamber pressure. The uncertainty of T_{sat} was estimated within 0.3°C. The turbine flow meter was calibrated. The uncertainty of the flow rate was 3% of reading.

RESULTS AND DISCUSSION

Experimental data expressing heat transfer characteristics, q'' vs. T_w-T_{sat}, for FC-72 and water are presented in the following figures. The nozzle pressure drop and the mass flow rate (kg/s) per nozzle are the operating conditions pertinent to the curves. Also presented in these figures are the saturation temperature, T_{sat}, in the spray chamber and the subcooling, T_{sat} - T_l, where T_l is the inlet liquid temperature.

Figure 3 shows the heat transfer characteristics with FC-72 at a nozzle pressure drop of 1.72 bar for the vertical cooling surface and the horizontal facing downward cooling surface for comparison. The results were related with three different spray saturation temperature levels. Each curve connected the data points for the same saturation temperature level (within a difference of 1.5°C). The subcooling, T_{sat}-T_l, was less than 6°C for all the data points. It may be seen that the slopes of the curves varied with the surface superheat and this implied a change in the importance of the nucleate boiling heat transfer with the surface superheat. As shown, the heat flux increased with T_{sat} for a given surface superheat. For a given T_{sat}, the heat transfer coefficient, $h=q''/(T_w-T_{sat})$, was slightly higher for the horizontal facing downward surface than for the vertical surface. This was attributed to the fact that in the case of the horizontal facing downward surface, the liquid could be more effectively discharged from the hot surface to the side channel with the aid of the gravitational force and therefore reducing the effective liquid film thickness on the hot surface. Figure 4 shows the heat transfer characteristics with FC-72 at a nozzle pressure drop of 1.72 bar for the vertical cooling surface and the horizontal facing upward cooling surface. For a given T_{sat}, the heat transfer coefficient was slightly lower (by 6%) for the horizontal facing upward surface than for the vertical surface because the effective liquid film thickness was greater for the former. Figure 5 shows the heat transfer characteristics with water for the vertical cooling surface. For a certain T_{sat}, the heat transfer coefficient was lower at a higher subcooling. At the same nozzle pressure drop, the heat transfer coefficient was higher at a higher T_{sat}.

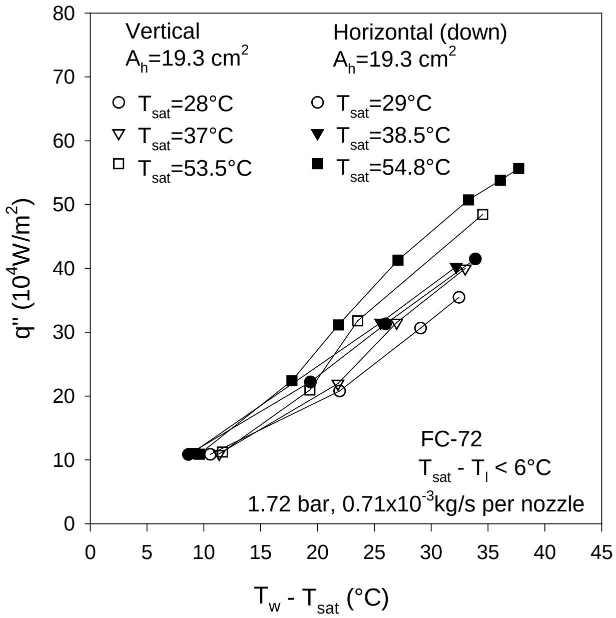

Figure 3. Heat transfer characteristics with FC-72 for the vertical cooling surface and horizontal facing downward cooling surface.

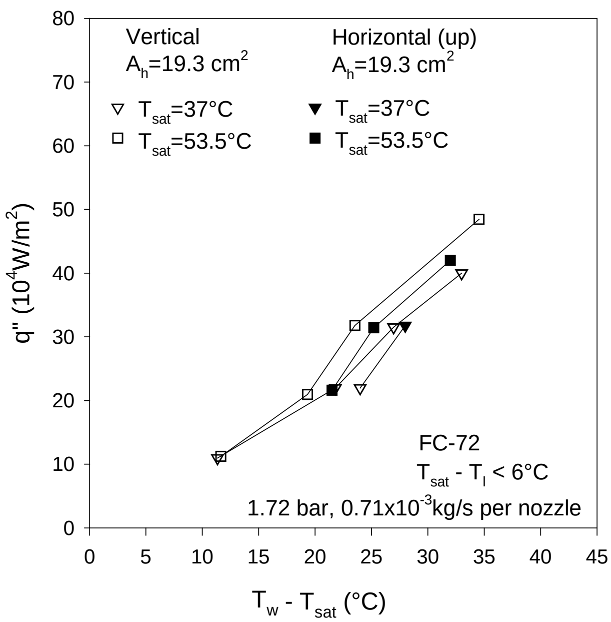

Figure 4. Heat transfer characteristics with FC-72 for the vertical cooling surface and horizontal facing upward cooling surface.

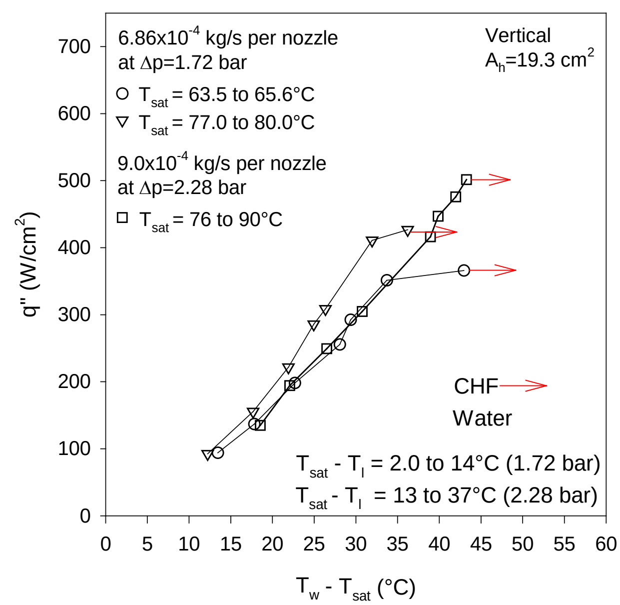

Figure 5. Heat transfer characteristics with water for the vertical cooling surface.

It was also observed during the experiment that using the ejector in the closed two-phase loop enhanced the capability of maintaining the two-phase fluid circulation. With the assistance of the ejector, the maximum spray pressure drop across the nozzle could be enhanced by 0.56 bar at critical heat fluxes (CHF). This increased CHF of the spray cooling by up to 16%.

Major spray cooling parameters are listed in Table 1 for comparison. The effectiveness of spray cooling at CHF was defined as the ratio of the heat that was actually removed at CHF to the total latent heat that could be removed by the spray and was written as

$$\eta_c = \frac{q''_c}{h_{fg} Q'' \rho_l}. \tag{1}$$

The heat transfer coefficient of the spray cooling was defined as

$$h = \frac{q''}{T_w - T_{sat}}. \tag{2}$$

The maximum heat transfer coefficient obtained for a given saturation temperature was denoted by h_{max}. The results of η_c, CHF and h_{max} as well as other parameters for the two working fluids are listed in Table 1. The results for the small cooling area, A_h = 2.0 cm^2, came from a previous 8-nozzle spray cooling study (Lin & Ponnappan, 2003). It was shown that η_c was much smaller for water than for FC-72 at the same Δp. It was exhibited from the variations in h_{max} with Δp that the spray heat transfer was enhanced with increasing the pressure drop in most cases. The heat transfer coefficient was much higher for water than for FC-72. The CHF values were lower for the large cooling area than for the small cooling area. Since the subcooling of the working fluids was controlled to be as small as possible, the effect of the subcooling on the heat transfer coefficient was precluded from the discussion of the results.

Table 1. Spray cooling parameters

Working Fluid	T_{sat} (°C)	Δp (bar)	CHF (W/cm^2)	η_c (-)	$h_{max}\times10^{-3}$ (W/m^{2}K)
FC-72 (small A_h)	54	1.03	65.0	0.324	16.7
		1.72	72.5	0.300	19.4
		2.41	78.5	0.282	20.7
		3.10	83.5	0.271	22.3
FC-72 (large A_h)	53	1.72	51.6	0.223	14.02
		3.10	60.0	0.256	15.3
Water* (small A_h)	70	1.03	>500	>0.116	84.2
		1.72	>500	>0.101	94.6
		2.41	>500	>0.088	97.8
Water (large A_h)	78	1.72	430	0.081	118
		2.28	500	0.102	116

* at 500 W/cm^2

CONCLUSIONS

- For a given spray saturation temperature, the thermal performance for the horizontal facing downward surface was slightly higher by 5.0% but the one for the horizontal facing upward surface was lower by 6% on the average than that for the vertical surface. This was attributed to the fact that the effective liquid film thickness on the cooling surface tended to decrease in sequence of (c), (b) and (a) orientations. It was believed that decreasing the effective liquid film thickness enhanced the probability for the droplets to touch the hot surface and, therefore, increased thermal performance.
- The use of the ejector stabilized the circulation of the two-phase flow.
- The spray cooling system with the large cooling surface area of 19.3 cm^2 reached CHF of 500 W/cm^2 for water as the working fluid. Compared with the water data of a small cooling surface area of 2.0 cm^2, the CHF values of the large area (19.3 cm^2) spray cooling were lower. However, the heat transfer coefficients were slightly higher for the large cooling surface than for the small cooling surface due to the proper design of the side channel for the large cooling surface.

NOMENCLATURE

A_h	=	cooling surface area, m^2
h	=	heat transfer coefficient, W/m^{2}K
h_{fg}	=	latent heat of vaporization, J/kg
p	=	pressure, N/m^2
q''	=	heat flux, W/cm^2
q''_c	=	critical heat flux, W/cm^2
T_l	=	inlet liquid temperature, °C
T_{sat}	=	spray saturation temperature, °C
T_w	=	cooling surface temperature, °C
Q''	=	volumetric flow rate per unit cooling area, m^3/m$^2\cdot$s
Δp	=	nozzle pressure drop, N/m^2
η_c	=	effectiveness of spray cooling at CHF
ρ_l	=	liquid density, kg/m^3

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