Jet Impingement Cooling

by Using FC-72

Jet impingement is one of the most efficient solutions for cooling hot electronic devices, as it can create a very high heat transfer rate on an impacting surface. Direct cooling of high-power chips, by using jet impingement via dielectric liquids, shows great potential even though its implementations are more complicated than traditional air and liquid impingement cooling methods. Unlike the conventional air and liquid impingement cooling which rely on heat sinks or cold plates, using dielectric liquids in intimate contact with silicon devices eliminates, deleterious effects of solid–solid interface resistances and thermal impedance created by thermal interfere materials.

The selection of a fluid for direct impingement cooling cannot be made solely on the basis of heat transfer characteristics. Chemical compatibility of the coolant with the chips and other packaging materials exposed to the liquid must be a primary consideration. There may be many coolants which can provide adequate cooling, but only a few will be chemically compatible with electronic cooling applications. Water is an example of a liquid which has very superior heat transfer characteristics, but it is generally unsuitable for direct cooling, due to its chemical characteristics. Fluorocarbon liquids (e.g. FC-72, FC-77, FC-87, etc.) are generally considered to be the most suitable liquids for direct immersion cooling, in spite of their poorer thermo- physical properties. As shown in Table 1, the thermal conductivity, specific heat and latent heat of fluorocarbon coolants are much lower than water [1]. These coolants are clear, and colorless per fluorinated liquids with a relatively high density and low viscosity. They also exhibit a high dielectric strength and a high volume resistivity. The boiling points for the fluorocarbon liquids, manufactured by the 3M Company, range from 30 to 253 °C.

PROPERTY	FC-87	FC-72	FC-77	H_2O
Boiling Point @ 1 Atm (°C)	30	56	97	100
Density x 10^{-3} (kg/m^3)	1.63	1.68	1.78	0.1
Specific Heat x 10^{-3} (Ws/kgK)	1.09	1.09	1.17	4.18
Thermal Conductivity (W/mK)	0.05	0.05	0.06	0.63
Dynamic Viscosity x10^4 (kg/ms)	4.20	4.50	4.50	8.55
Heat of Vaporization x10_L^{-4} (Ws/kg)	8.79	8.79	8.37	243.8
Surface Tension x10^3 (N/m)	8.90	8.50	8.00	58.9
Thermal Coefficient of Expansion x 10^3 (K^{-1})	1.60	1.60	1.40	0.20
Dielectric Constant	1.71	1.72	1.75	78.0

Table 1. Properties of Common Fluorocarbon Liquids and Water [1]

Many research initiatives have been conducted for microchannel and impingement cooling by using Fluorocarbon liquids. Meyer et al. [2] did experiments on the thermal performance of rectangular slot impingement jets, by using Fluorinert FC-72 as the working fluid. They investigated the effects of jet width, impingement velocity and inlet subcooling on the cooling ability of an array of three confined rectangular jets, in both single-phase and two-phase flow regions. Figure 1 shows the schematic of the flow loop they used to supply FC-72 to a jet-impingement test module. Figure 2 shows major components of the test module and flow moving pattern within the test module. According to the description of Meyer et al., the heater block was made from oxygen-free copper. The top 30 mm×30 mm surface of the copper block comprised the test surface of the module subjected to jet impingement. Three type K thermocouples were used to monitor the temperature of the test surface. The thermocouples were inserted 2.54 mm below the impingement surface, as shown in Figures 3(a) and (b). The operating conditions for FC-72 are summarized in Table 1. For each test, FC-72 was conditioned to enter the test module at either 10.6 ± 0.3 or 20.6 ±0.3°C subcooling temperature. Pump capacity set upper velocity limits of 3, 5, and 8 m/s for the 0.508, 0.254 and 0.127 mm jet plates, respectively.

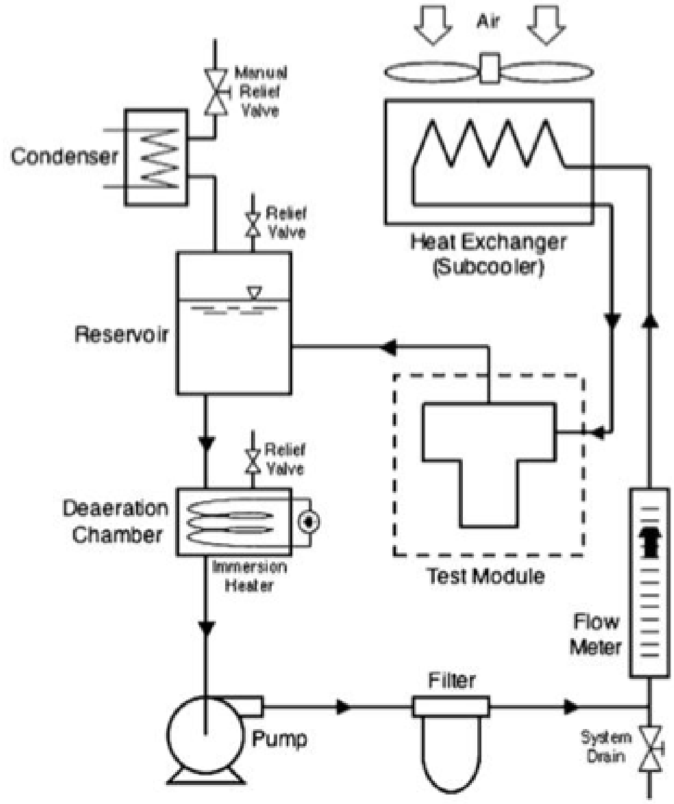

Figure 1. Schematics of Impingement Test Flow Loop [2]

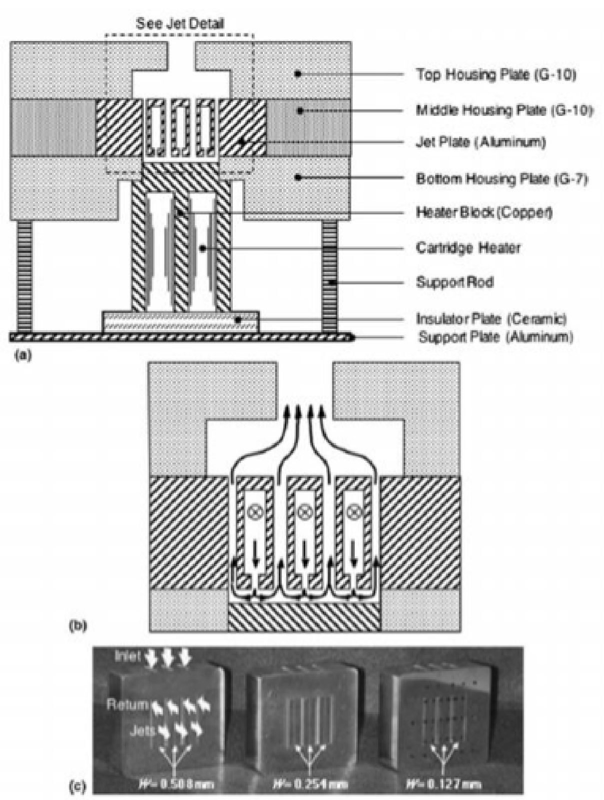

Figure 2. (a) Test Module Schematics (b) Fluid Pattern Inside Test Module (c) Photos of Underside of Jet Plates [2]

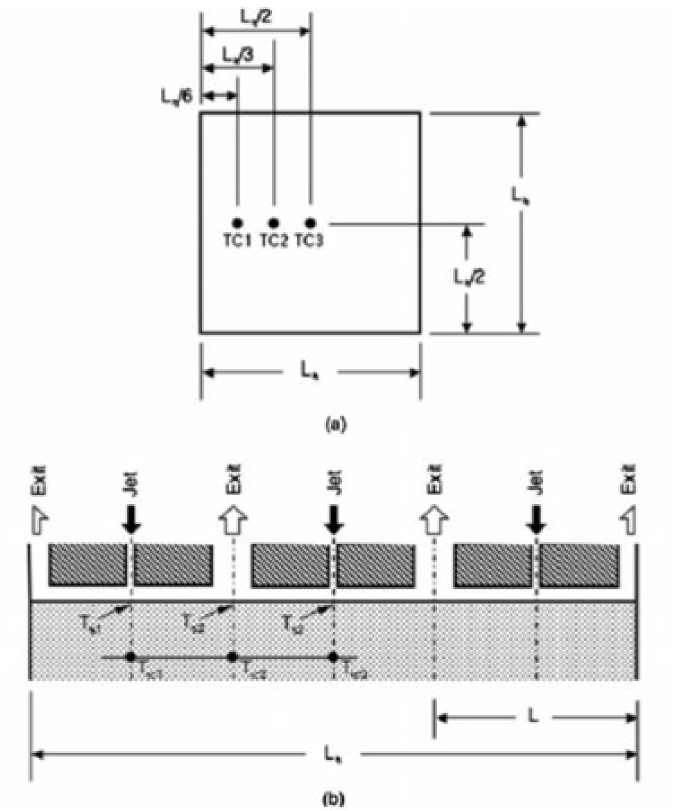

Figure 3. Thermocouple Locations Shown in (a) Top View and (b) Side View of Heater Block [2]

Jet width, W	0.127 mm	0.254 mm	0.508 mm
Inlet subcooling at CHF, ΔT_{sub}	10.6 ± 0.2 °C, 20.6 ± 0.2 °C	10.6 ± 0.2 °C, 20.6 ± 0.2 °C	10.6 ± 0.2 °C, 20.6 ± 0.2 °C
Jet velocity, U	2.0–8.0 m/s	1.0–5.0 m/s	1.0–3.0 m/s
Inlet pressure, p_{in}	1.09–1.51 × 10^5 N/m^2 (15.82–21.96 psi)	1.06–1.43 × 10^5 N/m^2 (15.31–20.68 psi)	1.11–1.88 × 10^5 N/m^2 (16.05–27.25 psi)
Outlet pressure, p_{out}	1.03–1.09 × 10^5 N/m^2 (14.91–15.84 psi)	1.04–1.14 × 10^5 N/m^2 (15.01–16.52 psi)	1.04–1.18 × 10^5 N/m^2 (15.08–17.12 psi)
Saturation temperature, T_{sat} (at p_{out})	57.1–58.9 °C	57.3–60.2 °C	57.4–61.2 °C

Table 2. Test Conditions for FC-72 Impingement Experiments [2]

Figures 4(a) and (b) show FC-72 boiling curves for the 0.127 mm jet for 10.6 and 20.6°C subcooling conditions, respectively. It is found that single-phase heat transfer prevailed over a broad range of surface temperatures. In this region, the heat flux increases linearly, corresponding to the increase of surface to fluid temperature difference. The single-phase heat transfer coefficient also increases with the increase of fluid velocity. The boiling region is evident in plots as the heat flux increases dramatically with a small increase of temperature difference. The critical heat flux (CHF) is directly related to fluid velocity. A higher velocity results in higher CHF. At 8 m/s and 20.6 °C subcooling conditions, the CHF can be as high as 139 W/cm^2. Figures 5(a) and (b) shows similar trends for the 0.508 mm jet. Both the single- phase heat transfer coefficient and CHF of the 0.508 mm jet are higher than those for the 0.127 mm jet for the same jet velocity. However, the 0.508mm jet has a much higher volumetric flow rate, due to its large cross section area.

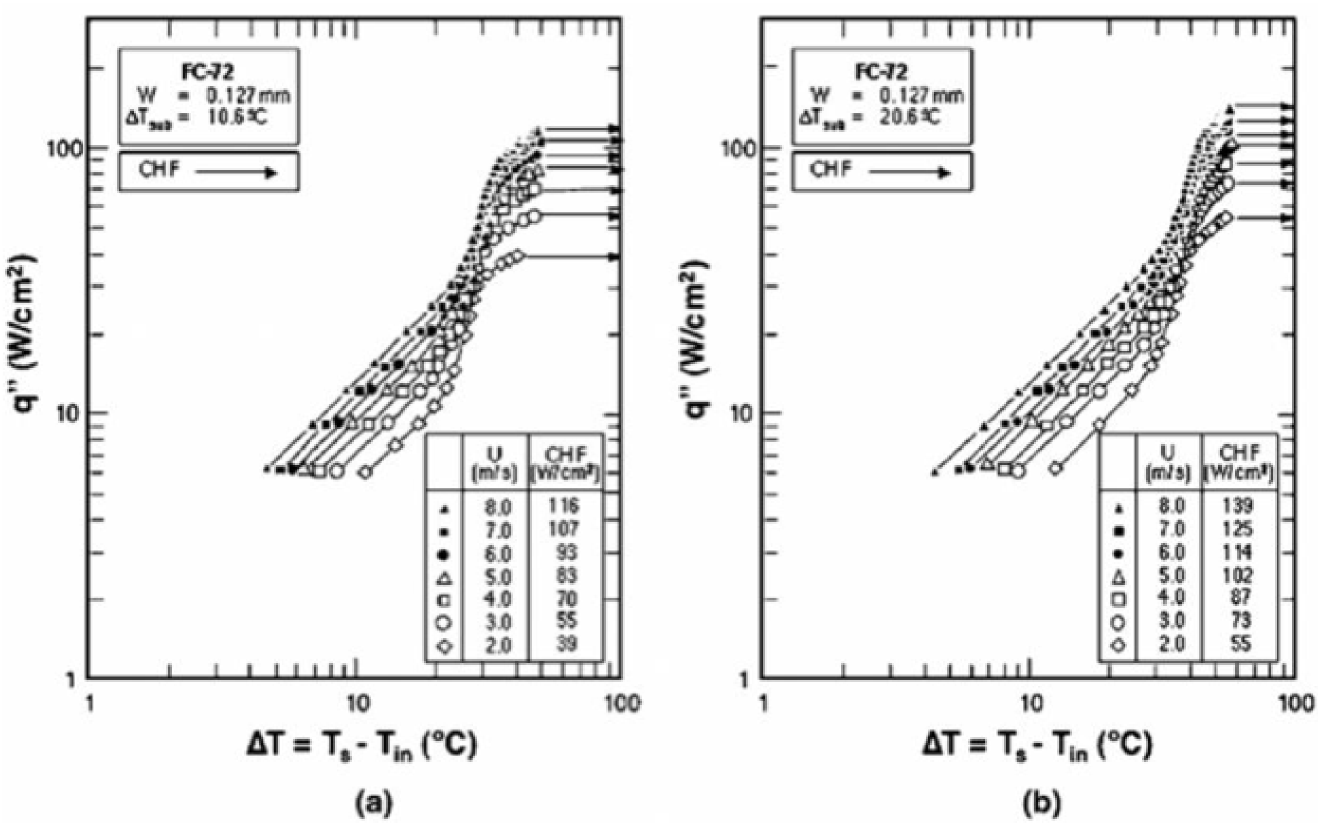

Figure 4. FC-72 Boiling Curves for 0.127 mm Jet with (a) 10.6 °C Subcooling and (b) 20.6 °C Subcooling [2]

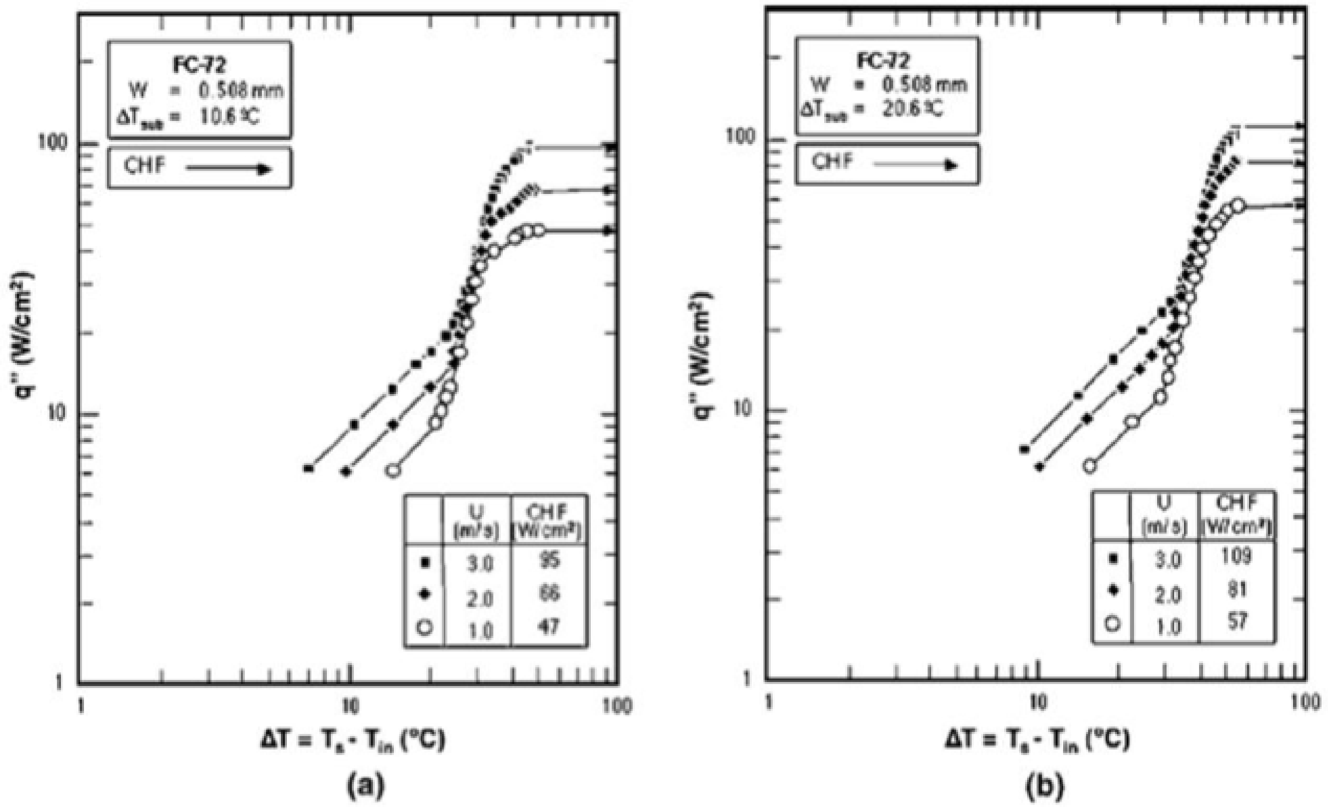

Figure 5. FC-72 boiling Curves for 0.508 mm Jet with (a) 10.6 °C Subcooling and (b) 20.6 °C Subcooling [2]

Fabbri et al. [3] experimentally studied the effect of the chip-to-orifice gap and orifice geometry on jet impingement performance. In their test, FC-72 was used as working fluid. The schematic of their test setup is shown in Figure 6.

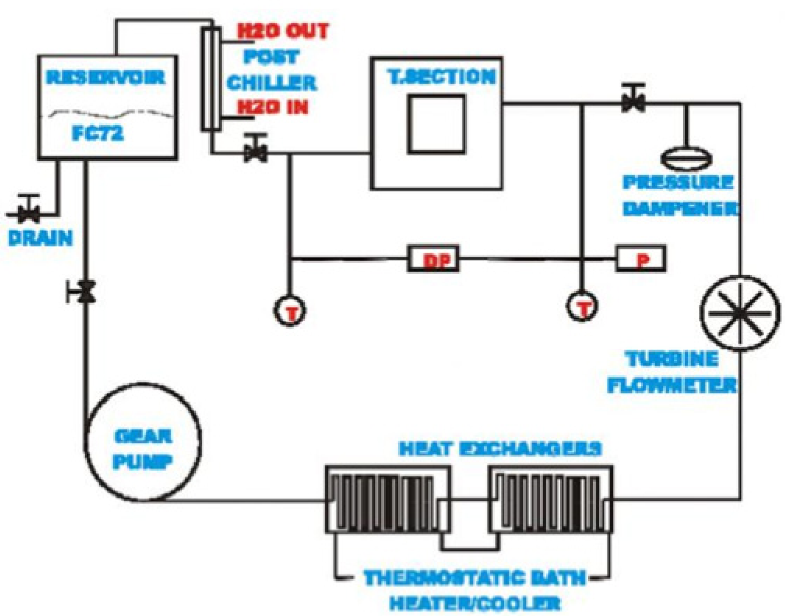

Figure 6. Test Setup Schematic [3]

The test section consisted of a support base made of aluminum, a PCB board carrying the heater chip, a spacer, the slot nozzle component and an aluminum manifold. The 20×20×1 mm^3 heater chip was made of silicon, with a 14×14 mm^2 NiCr thin-film resistive heater deposited on the back side together with five resistive temperature detectors. The schematic of the jet plate is shown in Figure 7 and the geometrical details of four jet arrays tested by Fabbri et al. are displayed in Table 3. On the jet plates, the outlet channels alternate with the inlet orifices, with each inlet being drained by two outlets and each outlet draining the flow from two jets, except for the outermost row of outlets. The outlet channels have been made larger than the inlet to reduce the flow velocity and thus prevent a large increase in the pressure drop when boiling occurs.

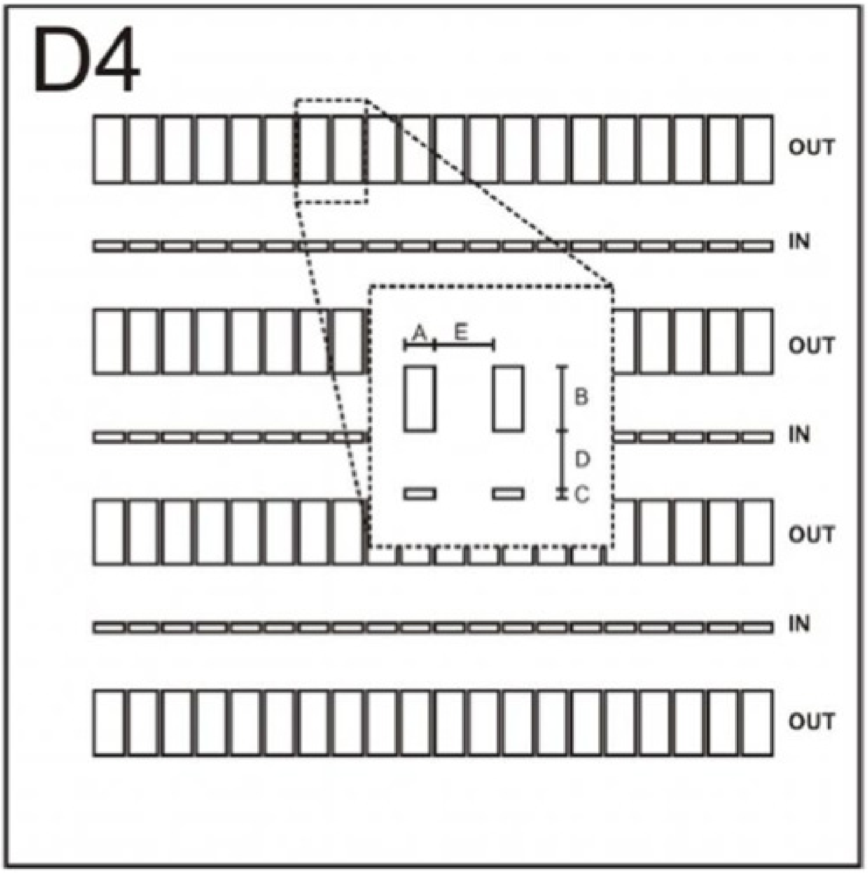

Figure 7. Nozzle Plate Schematic [3]

Design	A [mm]	B [mm]	C [mm]	D [mm]	E [mm]	# $Rows_{IN}$	# $Rows_{OUT}$	# Ch./row	Dh_{IN} [mm]	Dh_{OUT} [mm]	Total $Area_{IN}$ [mm]	Total $Area_{OUT}$ [mm]
D1	0.703	0.7	0.147	0.775	0.99	6	7	20	0.243	0.701	12.40	68.89
D2	0.697	0.903	0.194	0.649	0.93	6	7	20	0.304	0.787	16.23	88.11
D3	0.7	0.9	0.099	0.703	0.102	6	7	20	0.173	0.788	8.32	88.20
D4	0.7	1.501	0.199	1.376	0.99	3	4	20	0.310	0.955	8.36	84.06

Table 3. Nozzle Plate Details [3]

Figure 8 shows the heat dissipation of a nozzle plate with configuration D4 as a function of the difference between the maximum chip temperature (measured at chip center) and the inlet liquid temperature for constant flow rate and as a function of T_{liq}. Clearly, the heat flux increases linearly with the increase of $T_{chip\ max}$-T_{liq} in the single phase region. In the boiling region, the jet dissipates more heat flux with a smaller temperature increase.

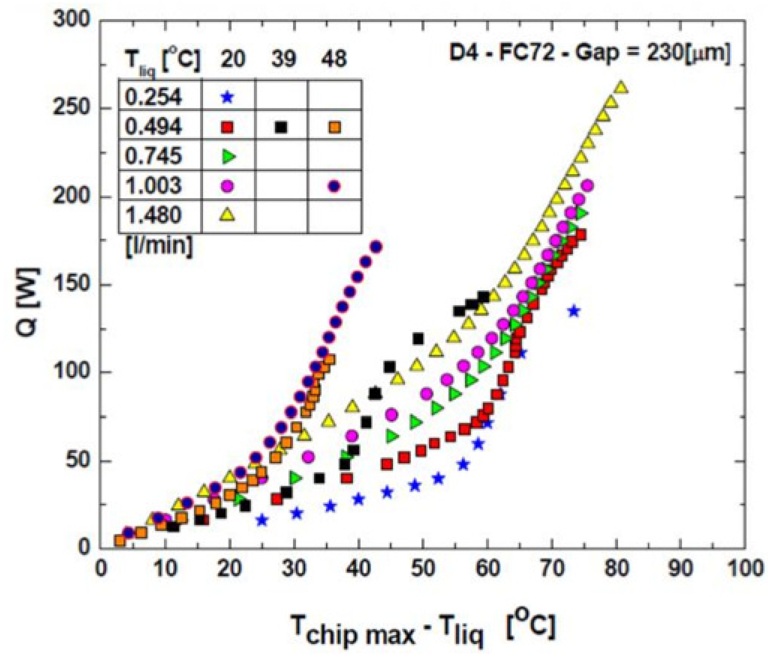

Figure 8. Effect of Different T_{liq} and Liquid Flow Rate on the Module Performance for Configuration D4 [3]

The effect of the gap between the nozzle plate and the heater on the module performance is illustrated in Figure 9, for configuration D4 and T_{liq}=20°C. Fabbri et al.[3] found out that a smaller gap yields a better heat transfer at low flow rates. And, there is no significant difference between gap sizes at a flow rate of 1.5 l/min. This is expected, as the penetration depth of a jet increases with increasing velocity.

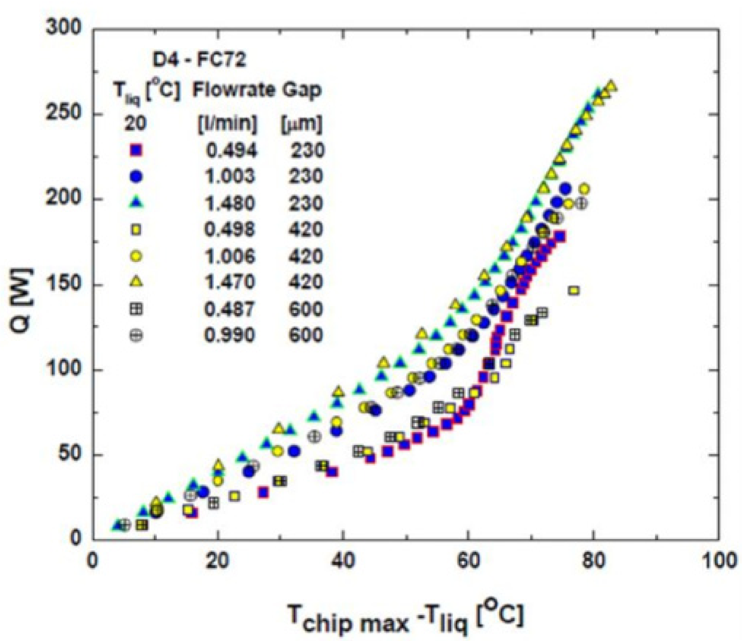

Figure 9. Gap Effect for Configuration D4 and T_{liq}=20°C [3]

Figures 10 and 11 show the comparison of different nozzle configurations at different liquid volumetric flow rates. From these figures, it is evident that there are no great differences in the performance of the four configurations. The configuration D3, which has the smallest inlet hydraulic diameter and the thinnest jets, dissipates less heat than other configurations at 1.5 ml/min flow rate.

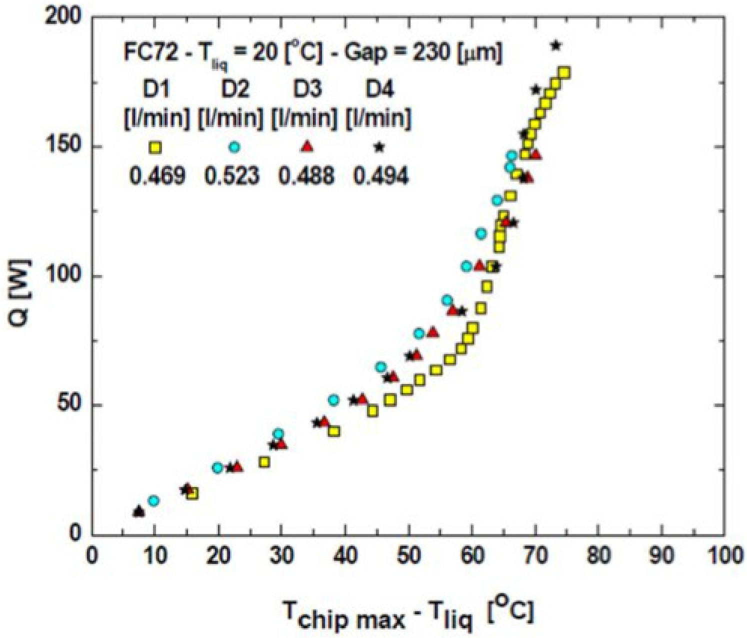

Figure 10. Performance of Different Configurations mv=0.5 l/min and T_{liq}=20°C [3]

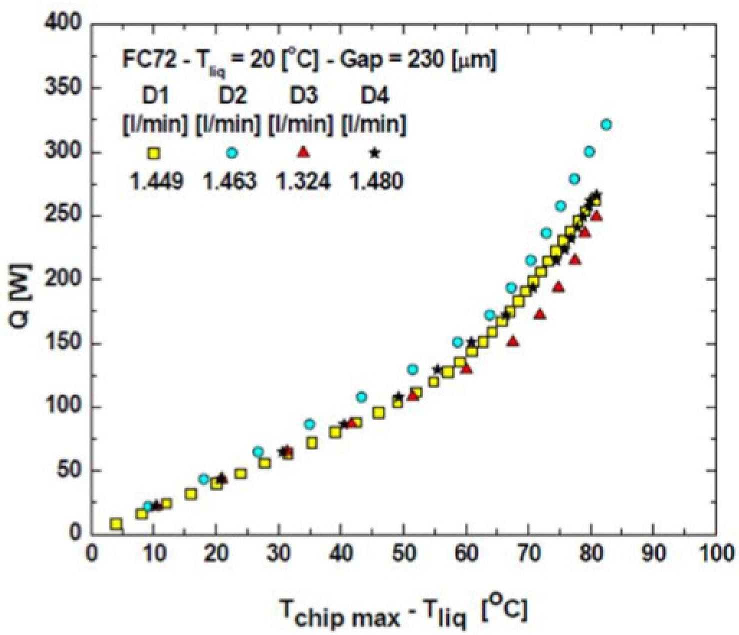

Figure 11. Performance of Different Configurations mv=1.5 l/min and T_{liq}=20°C [3]

The experiments conducted by Meyer et al. [2] and Fabbri et al. [3] show that FC-72 is suitable for direct impingement cooling of electronic devices. In the single-phase region, the heat transfer coefficient for FC-72 increases with increases in impingement velocity and jet width. Meyer et al. found that increases in jet velocity, jet width and subcooling broaden the single-phase region, preceding the commencement of boiling. Within the nucleate boiling region, data for different jet widths and velocities for a given subcooling tend to converge. Fabbri et al [3] reach similar conclusions through their tests. In both experiments, a cooling heat flux of over 90 W/cm^2 was achieved by using FC-72 jet impingement with moderate pressure drop. This demonstrates that the two-phase jet impingement direct cooling is a viable method to cool high heat flux devices. However, more experiments are needed to evaluate the performance with a given pump and pressure head. In this case, the geometry of the nozzle changes the pressure drop and hence the volumetric flow rate and velocity.

References:

1. Danielson, R., Tousignant, L., and Bar-Cohen, A., Saturated Pool Boiling Characteristics of Commercially Available Perfluorinated Liquids, Proc. of ASME/JSME Thermal Engineering Joint Conference, 1987.

2. Meyer, M. T., Mudawar, I., Boyack, C. E., and Hale, C. A., Single-phase and Two-phase Cooling with An Array of Rectangular Jets, International Journal of Heat and Mass Transfer 49 (2006) 17–29.

3. Fabbri, M., Wetter, A., Mayer, B., Brunschwiler, T., Michel, B., Rothuizen, H., Linderman, R., and Kloter, U., Microchip Cooling Module Based on FC72 Slot Jet Arrays without Cross-Flow, Semiconductor Thermal Measurement and Management Symposium, 2006.