Journal of Advances in Science and Technology

Vol. VI, Issue No. XII, February-2014, ISSN 2230-9659

CONVECTION AND FLOW BOILING IN MICROGAPS AND POROUS FOAM COOLERS

AN INTERNATIONALLY INDEXED PEER REVIEWED & REFEREED JOURNAL

www.ignited.in

Journal of Advances in Science and Technology
Vol. VI, Issue No. XII, February-2014, ISSN 2230-9659

Convection and Flow Boiling In Microgaps and Porous Foam Coolers

Suresh Pittala

Research Scholar, University Himalayan, Arunachal Pradesh

Abstract – The flow regime progression in such a micro gap channel is shown to be predicted by the traditional flow regime maps. Moreover, annular flow is shown to be the dominant regime for this thermal transport configuration and to grow in importance as the channel diameter decreases. An open and foam-filled micro gap cooler, providing direct liquid cooling for a simulated electronic/photonic component and which eliminates the problematic thermal resistance of the commonly-used thermal interface material (TIM), is examined. The single phase heat transfer and pressure drop results of water are used to validate a detailed numerical model and, together with the convective FC-72 data, establish a baseline for micro gap cooler performance.

Keywords: Convection, Boiling, Micro Gaps, Foam Coolers

---------------------------♦-----------------------------

INTRODUCTION

The two-phase heat transfer characteristics of FC-72 are examined at various micro gap dimensions, heat fluxes, and mass fluxes and the results are projected onto a flow regime map. Infrared (IR) thermograph is used to explore the two-phase characteristic of FC-72 inside the channel instantaneously. Also the single and two-phase heat transfer and pressure drop of porous metal foam which can enhance the cooling capability of low conductive fluid are studied and compared with the performance of the open channel micro gap cooler in terms of volumetric heat transfer rate and required pumping power. The single-phase experimental results were in good agreement (within 10% error) with classical correlation of single-phase heat transfer coefficient and pressure drop in micro single gap channel with heat transfer coefficients as high as 23 kW/m2- K at 260 μm gap with water and 5 kW/m2-K at 110 μm gap with FC-72. Annular flow was found to dominate the two-phase behavior in the open channel yielding FC- 72 heat transfer coefficients as high as 10 kW/m2-K at 110 μm gap channel. The single-phase pressure drop and heat transfer coefficient experimental results are compared with existing correlations and achieved 10 kW/m2-K of heat transfer coefficient at 95% porosity and 20PPI with water and 2.85 kW/m2-K with FC-72 at the same configuration. For the two-phase flow boiling, it is found that large pore size provides better cooling capability.

REVIEW OF LITERATURE:

Local micro gap heat transfer coefficients

In a parallel study (Rahim et al 2012), using Intel's 11mm x 14mm Thermal Test Vehicle to simulate an actual silicon die, local heat transfer coefficients were obtained via the inverse calculation procedure described above and compared to the predictions of two classical correlations by Chen and Shah. As can be seen in Table 1, the average discrepancy between local h's and the Chen correlation for Annular flow is approximately 23% and results for Intermittent flow can be predicted to within 33% by both correlations. The agreement between data and correlations shows some dependence on gap size, with the larger gap of 500 micron showing 20% agreement with Chen and deteriorating as the gap size get smaller to 54% for the 100 micron channel. The discrepancy was moderately smaller for FC-87 data than HFE-7100 data, with the FC-87 data showing discrepancy of 26% and 41% with Chen and Shah, respectively, while the HFE-7100 data was found to show a discrepancy of 35% and 53% with same correlations.

Local Heat Transfer Coefficients Comparison		Number of Points	Discrepancy %	
			Chen	Shah
All Data		6210	30	47
Fluid	hfe-7100	3096	35	53
	fc-87	3114	26	41
Gapsize (Micron)	100	1404	54	85
	200	1926	28	52
	500	2880	20	25
Flow Regime	Intermittent	4338	33	33
	Annular	1872	23	78

Operational micro gap effects

Under operational conditions, electronic modules may be subjected to orientation changes and may experience substantial random vibration from the underlying structure or from other equipment nearby. In Figure 11 the sensitivity of the local microgap heat transfer coefficient to a change in orientation, from horizontal to vertical, is displayed for FC-72 flowing in a 210 micron high channel at two different mass fluxes and heat fluxes. In Fig 11a, it may be seen that at the low mass flux of 195kg/m2-s and a heat flux of 24W/cm2, the horizontal orientation yields approximately 15% higher microgap heat transfer coefficients, consistently along the channel. However, at an increased mass flux of 488 kg/m2-s, even in the presence of a 60W/cm2 heat flux, there are no statistically significant variations of the local heat transfer coefficients with orientation.

- **Single-Phase Heat Transfer**

The thermal performance of a micro gap cooler is evaluated with the

Dimensionless Nusselt number as given by

$Nu = hD_{h/}\, k$

Where k is the thermal conductivity of the liquid, h D is the hydraulic diameter, and h is the heat transfer coefficient. The heat transfer coefficient is defined as

$h= q/A\Delta T$ (1.1)

where ΔT is the temperature difference between the heated surface (copper block) and coolant and q is heat transfer rate to the coolant. The heat transfer rate can be determined from an energy balance on the channel, i.e.

$q = m\&C\,(T_{outlet} - T_{inlet})$ (1.2)

where $m\&$ is the mass flow rate of the coolant.

From the above equations, the empirical Nusselt number can be calculated by the following expression, using the experimental data.

$Nu=MCp(T_{outlet}-T_{inlet})Dh/\, A\, \Delta T\, K$

In their classic textbook, [Kays and Crawford] discuss many of the available Nu equations for uniform surface temperature and uniform surface heat flux boundary conditions, applied to circular tubes, circular tube annuli, and rectangular tubes and covering the laminar flow of fluids of various Prand tl number for both fully developed and thermally-developing flow. Therefore, single-phase, laminar flow heat transfer can be predicted very well in most cases. [Mercer et al.], performed experiments with a 12.7 mm (0.5") high parallel plate, copper channel with air flow and correlated their experiment data in terms of the thermally-developing length x*.

The local and mean heat transfer coefficients for isoflux,

Two-phase-

- **Pool Boiling**

The heater attached copper base block was immersed into the fluid of FC-72 and the heat flux increased. The saturation temperature was set as 56°C which was the boiling temperature of FC-72 at 1 atm. The boiling curve can be separated into two-regimes; one is natural convection, which occurred until slightly above the saturation temperature, and the other regime which occurred after the natural convection regime. The critical heat flux of this experiment was 37.5 W/cm2.

- **Flow Boiling**

The two-phase flow boiling experiment was performed with mass flux varying from 56 kg/m2-s to 1270 kg/m2-s and at an input power of up to 25W. The sub cooled liquid was supplied, as previously described, from the pump to the micro gap, parallel plate channel.

- **Porous Foam Channel**

Porous foams are an attractive way to enhance convective heat transfer in tubes and channels due to the foam's high ratio of surface area/ volume. Moreover, the dispersion conductivity which is caused by the recirculation, mixing, and "disturbance" of the flow by the porous matrix further enhances the heat transfer rate.

The consequent relative insensitivity of channel heat transfer to the thermal conductivity of the fluid makes porous foams an attractive candidate for heat exchange with low conductivity fluids such as the dielectric fluids.

CONCLUSION:

In this paper we found that In the analysis of the characteristic of the porous foam channel in the single phase and two phase heat transfer, a heat transfer coefficient of 10 kW/m2-K was achieved with 95% porosity and 20 PPI with water and a heat transfer coefficient of 2.85 kW/m2-K was achieved at the same configuration of foam with FC-72 for the single phase.

REFERENCES:

1. Rahim, E., Bar-Cohen, A., and Ali, A., 2012, "Two-Phase Microgap Cooling of a Thermally-Simulated Microprocessor Chip," Proceedings, IEEE ITHERM Conference, San Diego, Ca.

2. Bar-Cohen, J. Sheehan, A. Elbaz, C. Holloway, **Thin Film Evaporative Cooling in Microgap Channels,** HEFAT2012 Keynote Extended Summary ABC 5/14/12

3. Shah, R. K. and London, A.L., 1978, “Laminar Flow Forced Convection in Ducts”, Supplement 1 to Advances in Heat Transfer, Academics, New York

4. Kays, W. M. and Crawford, C., 1966, “Convective Heat and Mass Transfer”, Mcgraw-Hill, New York

5. Mercer, W., Pearce, W. M., Hitchcock, J.E., 1967, “Laminar Forced Convection in the Entrance Region between Parallel Flat Plates”, J. Of Heat Transfer, pp. 251-257

6. Dae Whan Kim, convection and flow boiling in micro gaps and porous foam coolers, 2007