Experimental Investigation on Self-similar Heat Sinks for Liquid Cooled Electronics

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Abstract The high heat transfer coefficients in microchannels are attractive for direct cooling of electronic systems requiring high heat-flux removal. In this work we are presenting the results of a study on self-similar heat sinks for liquid cooled electronics, made from copper, designed for industrial application and for large scale production. The internal structures, where the most part of the active cooling takes place, have been designed in order to achieve high heat transfer coefficients.
As it is almost impossible to validate the design and describe the flow characteristics inside the device via analytical solutions, a well known numerical code was employed to have an insight of the thermal-fluid distributions. It is clear from the simulation that even if copper is characterized by a high thermal conductivity, most of the heat is removed in the overflow-structure, on the side of the device adjacent to the source of heat.
This paper attempts to critically analyse a comprehensive list of data as well as plots in order to illustrate the significant characteristics of this type of device.

Keywords: microchannel, developing boundary layer, heat sink, liquid cooling

1. Introduction

In electronic devices, thermal management issues play an increasingly prominent role in microelectronic system design. The constraints on heat removal are a major factor limiting the performance of a microelectronic system. The use of microchannels as a viable cooling solution was first proposed 30 years ago by Tuckerman and Pease [1]. They showed that a single layer microchannel etched directly on a silicon wafer is highly effective for dissipating heat. Using water as a working fluid they demonstrated that these microchannels can remove up to 790W/cm^2 of heat.
The high heat transfer coefficients in microchannels are attractive for direct cooling of electronic systems requiring high heat flux removal. The effectiveness of a microchannel device for high heat flux cooling lies in its increased heat transfer coefficient and in a large surface area to volume ratio. The effectiveness of micro heat exchangers has been widely implemented in different sectors of research and industry [2]. State of the art micro-scale convective heat transfer techniques are presented for use in heat sinks.
In chemistry and biological sciences microchannels are used in various microsystems such as micro heat sinks and microreactors, because of their different superior performances compared to conventional size devices.
The recent attention on micro devices has favoured not only the research in the thermal-fluid-dynamics, but also the field of micro-fabrication techniques, i.e. among others this includes hot-embossing, lithography, etching, micro-mechanical machining and diffusion-bonding.
Several researchers have investigated the possibility of implementing different shapes for the microchannels, ranging from rectangular, to semi-circular, to triangular and so on, both from a fluid-dynamic as well as a manufacturing point of view [2]. Even though, many have been oriented to using simple,

easily obtainable shapes.
In this work we are presenting the results of a study on self-similar heat sinks for liquid cooled electronics, made from copper, designed for industrial application and for large scale production. Here self-similarity refers to the fact that there is a certain similarity (or pattern) of the substructures compared with the overall structure. In fact, as per Figure 4, a main channel drives the water to a set of sub-channels and then from these to the overflow structures (microchannels, Figures 1 and 2). As it is almost impossible to validate the design and describe the flow characteristics inside the device via analytical solutions, a well known numerical code was employed to have an insight of the thermal-fluid distributions. It is clear from the simulation that even if copper is characterized by a high thermal conductivity, most of the heat is removed, in the overflow-structure, on the side of the device adjacent to the source of heat.
This paper attempts to critically analyse a comprehensive list of data as well as plots in order to illustrate the significant characteristics of this type of device.

2. Physical considerations and manufacturing

The internal structures where the most part of the active cooling takes place has been designed in order to achieve high heat transfer coefficients. Remarkably, using short overflow structures (identified as microchannels) it is possible to have a thermal and hydrodynamic developing flow where high thermal gradients are present, in fact as it has been pointed out by different authors [3, 4, 5] reducing the flow length and increasing the number of passages is advantageous in limiting the pressure drop and increasing the Nusselt number for laminar regimes [6].
Considering the energy equations relative to a 2D boundary layer and 1D heat conduction re-arranged as shown in [7] can provide an inside perspective of the manner in which heat is exchanged in a microchannel. In fact, a fundamental parameter in this equation results in the product of the velocity vector field and the gradient of the temperature. Reducing the intersection angle between these two quantities or maximizing their product, will lead to an enhancement of the convective heat transfer. This can be achieved having a non-developed flow, constant heat flux along the walls and short microchannels.
The fact that the flow is developing has been established using the correlations available in [8] and from a numerical point of view, using the well known Fluent package. Both have shown that for the cases considered in this work the flow is still within the developing region. This has a great advantage as due to the high gradients (thermal and velocity) from the energy equation the heat transfer is dramatically increased.
The issue arising when considering short microchannels is that they considerably reduce the heat exchange surface. To avoid this, the microchannels have been disposed as shown in figure 2, resulting in a relatively high exchange area, in this way compensating the smaller area available for a single short microchannel.
Figures 1 and 2 provide insight on the flow arrangement: it is quite clear that an analytical description of the flow remains elusive due to the 3-dimensional flow path. This is one of the reasons why a numerical study is necessary.

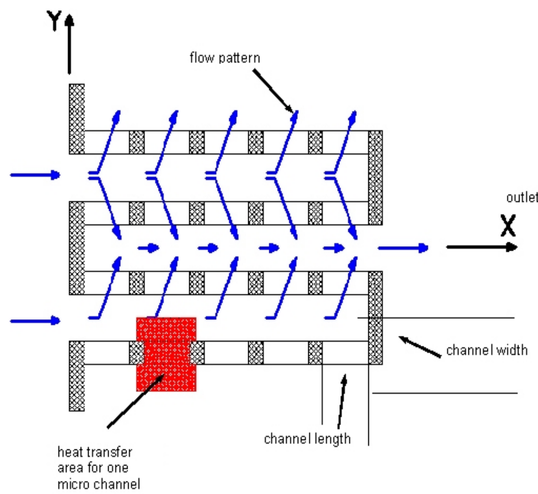

Figure 1: Schematic of the microstructure arrangement and the flow pattern

The devices and all the microstructures were manufactured in copper by means of mechanical precision machining. The overflow structures (namely the microchannels) were manufactured on the back side of the cover plate, subsequently the cover plate was connected with the main body by diffusion bonding.

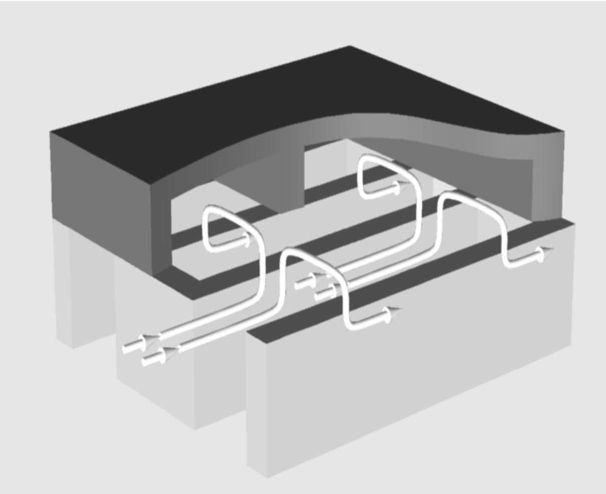

Figure 2: Sketch of the microstructure arrangement and flow pattern in the microstructures

Figure 3: Microstructure device developed at IMVT

As can be found in literature ([1], [2]), this technique has been demonstrated to be an excellent way to connect different layers (in this case of copper), resulting in a device with very few defects. Given the considerable overall dimensions of the devices (length 140mm, height 22mm and width 55mm), the heat exchanger area to be used in calculating the heat flux could be an issue, as not the entire device is active for cooling. Specifically, the question is when calculating the heat flux is the area to be considered the surface separating the device from the heating block or only the actual surface area of the active part (namely the microchannels)? Approaching these considerations, both the heat fluxes are given, as well as some explanations as to why the latter approach is preferred. Hence, a maximum and a minimum value are considered ranging the heat flux from approximately a minimum of 200W/cm^2 (for the widest area considered) up to more than 700W/cm^2 (for the tiniest area considered), the real value probably lying between these two values. An estimation of the heat losses in other parts of the device is given.

The micro-structured device is mainly divided in 3 sub-regions (as per Figure 4) and each of these is similar to what is shown in figures 1 and 2. The total number of overflow structures in the whole device is $N_{microchannels}$=3300, where the dimensions of a single microchannel are: width=0.8mm, height=0.1mm and length=0.2mm.

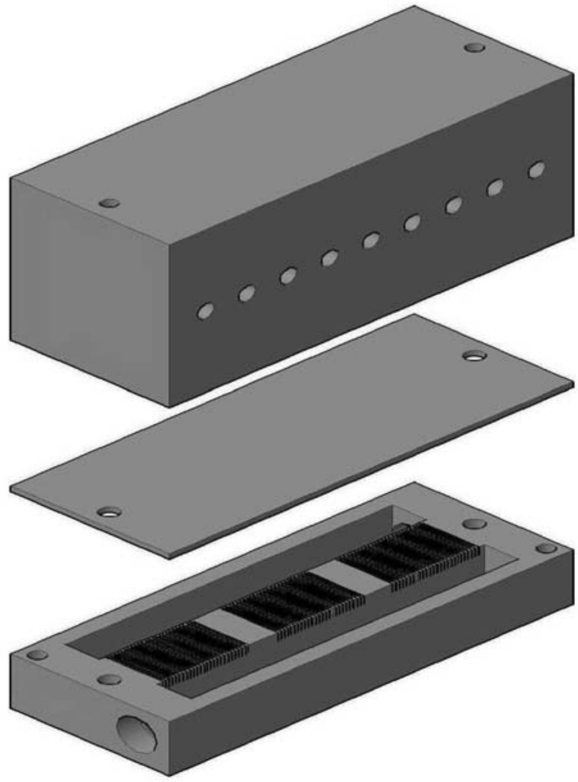

Figure 4: Exploded view of the microstructure device with the cover plate and the heating block

For this series of experiments, three devices (namely D1, D2 and D3) were manufactured, with self similar characteristics (same geometry and manufacturing process).

3. Experimental apparatus

In Figure 5 it is possible to see the test rig arrangement for the test series described in this paper.

Therefore, for the purpose of these experiments, some assumptions are needed:

- constant heat flux along the length of the overflow structures (microchannels);
- heat losses from the casing are neglected;
- the coolant flow is steady and incompressible.

In these experiments de-ionized water was used as a working fluid in a closed loop. A data acquisition system (DAS) was used to acquire the data and all measurements were taken in steady state conditions.

The mass flow rate has been varied from 300kg/h to almost 1000kg/h resulting in a pressure drop from approximately 45mbar to approximately 400mbar and the heat supplied was varied from approximately 600W up to approximately 6000W.

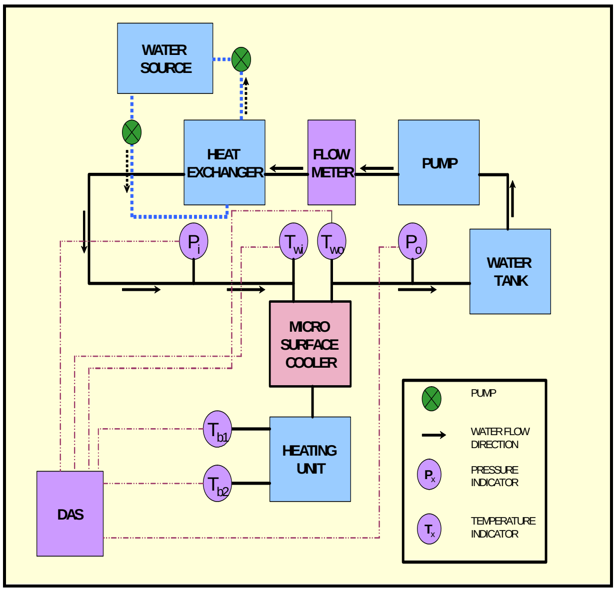

Figure 5: test rig arrangement

The water temperature at the inlet of the device was carefully controlled and kept at a constant value of 65°C. Thermoresistances Pt-100 were placed in the heating block, close to the contact surface with the microcooler to monitor temperature distributions.

4. Results

The experiments were conducted for a variety of conditions, in particular 5 different conditions regarding the thermal power were applied and 4 different settings for the mass flow rate were used.

One characteristic of flows in micro devices is the low Reynolds number. The range of the Reynolds number based on the hydraulic diameter at the inlet of the device is between Re_D=60 and Re_D=200, this will ensure a laminar flow in the main channels. For the overflow structures, the local Reynolds number ranges from Re_d=40 to Re_d=140 and also here the flow remains laminar. This is not surprising, in fact microfluidic devices employ fluids with Reynolds numbers that are small enough for inertial effects to be irrelevant [9].

When calculating the heat flux, the area where the heat is removed plays an important role. Ideally the whole surface of the heat exchanger contributes to the dissipation of the heat coming from the heating block, but not all with the same weight. Besides the quantity of heat rejected to the environment can be neglected being in the order of a few watts. Additionally, the 2 main channels that drive the water in and out of the 3 sub-structures, dissipate a certain amount of heat. The estimated thermal power dissipated is of the order of 20W for the lowest power supplied (approximately 600W), and 200W for the highest power supplied (approximately 6000W), that means something in the range of 3% for all the different heat loads.

Comparing different results especially for electronics cooling, it is interesting to plot the heat flux versus the temperature difference ΔT between the surface (in this case the separation surface between the heating block and the device) and the water at the inlet of the device.

The results, as per Figure 6 and Figure 7, show that increasing the mass flow has almost no appreciable effect on the temperature at the surface.

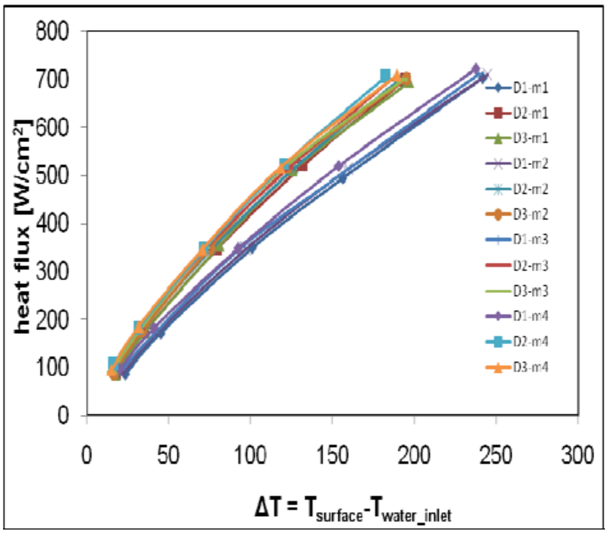

Figure 6: heat flux for the minimum area considered compared to the difference in temperature between the surface and the inlet water

In fact, from a mass flow rate of 300kg/h up to approximately 1000kg/h and for the same heat load, the decrease of ΔT is in the range of 5°C. The discrepancy in the results from device D1 to the other devices D2 and D3, can most probably be ascribed to a non perfect contact between the surface of the heating block and the micro device, as the contact pressure and the quality of the surface influence the contact resistance. In particular, this is evident in Figure 6 and Figure 7, as being the inlet temperature of the water constant, the temperature at the separation surface is higher, indicating a higher thermal resistance.

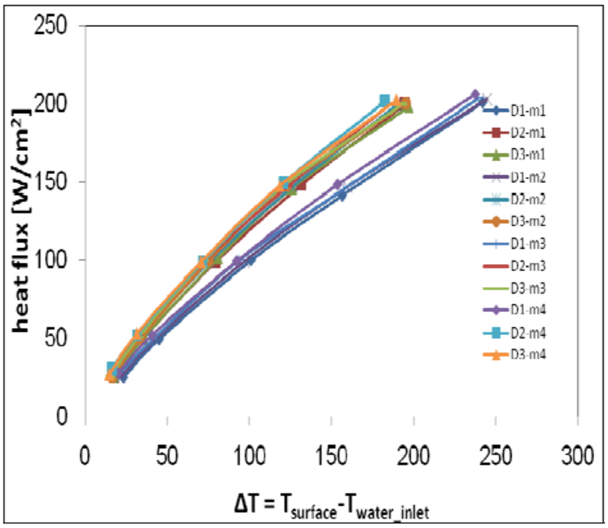

Figure 7: heat flux for the maximum area considered compared to the difference in temperature between the surface and the inlet water

As the inlet temperature was 65°C and the surface temperatures for D1, D2 and D3 ranged from 250°C to 300°C approximately, this implies that the temperature at the surface of the microchannels was still sufficiently high to cause boiling, but the inlet pressure was approximately 8bar and this is sufficient to suppress boiling. Comparing Figure 6 with Figure 7 it is evident that the heat flux changes dramatically, due to the choice of exchanging area. Giving a non-unique definition of the latter can present issues from an industrial point of view.

The pressure drop shown in Figure 8 shows a small difference for device D3 with respect to the other two. This is probably a consequence of some occlusions, possibly caused by imperfections or defects in the manufacturing process of the microchannels; trapped gas bubbles are improbable as care was taken to degasify the whole test rig as well as the surface cooler. Nevertheless, the pressure drop is remarkably low, for this type of device, even at the maximum mass flow rate of approximately 1000kg/h, thanks to the limited length of the overflow structures.

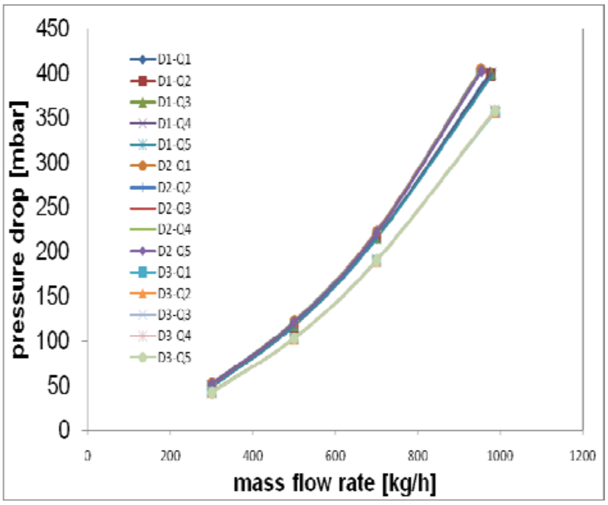

Figure 8: pressure drop against the mass flow rate

In order to give consistency to our research, some simulations were conducted with a standard CFD package.

Simulations were obtained for an entire row of microchannels, but here only the results relative to one single microchannel are shown, taken as representative for the thermo-fluid behaviour of the device. By analysing the velocity distribution the complex 3D pattern of the flow distribution is evident. In Figure 9, the y-axis indicates the direction along the length of the microchannel and the plane x-z results as a normal cross-section of the microchannel, thus, case a) was located 0.02mm after the microchannel entrance, b) in the centre of the microchannel and c) 0.18mm from the entrance and near the exit.

In this set of simulations no conjugate effects were considered as we focussed mainly on the hydrodynamic behaviour of the fluid and also because the computational capabilities were limited.

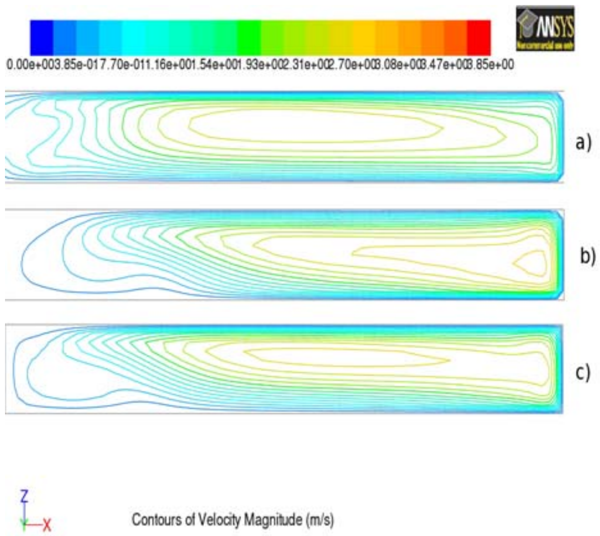

Figure 9: contour of the cross section velocity at 3 different positions throughout the length of the microchannel; a) 0.02mm; b) 0.10mm; c) 0.18mm

Figure 9 and Figure 10 together, indicate that the hydrodynamic boundary layer is still developing, giving confirmation to the previous considerations regarding the enhancement of the convective heat transfer.

The resulting velocity field is non-equally distributed, mostly because of the sudden variation in the direction due to the specific geometry of the device.

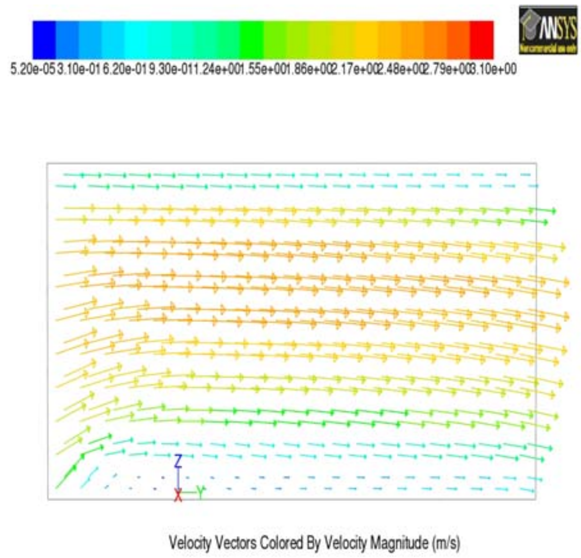

Figure 10: velocity field in a section along the length in the centre of the microchannel

For the temperature distributions, in Figures 11 and 12 it is clearly observed that even the thermal boundary layer is still developing, making the velocity and the temperature gradient in good synergy [7].

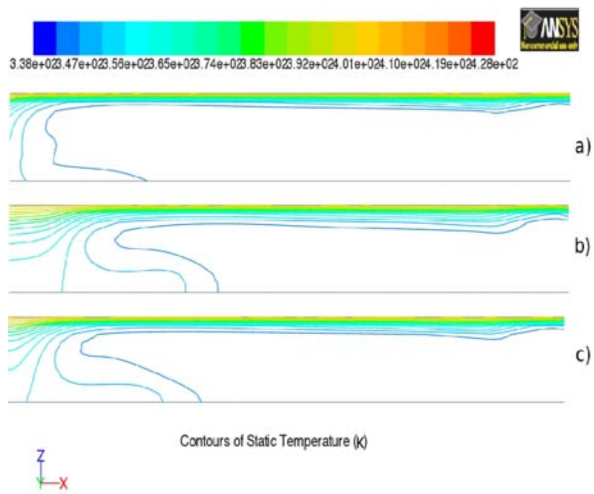

Figure 11: contour of the cross section temperature at 3 different positions throughout the length of the microchannel; a) 0.02mm; b) 0.10mm; c) 0.18mm

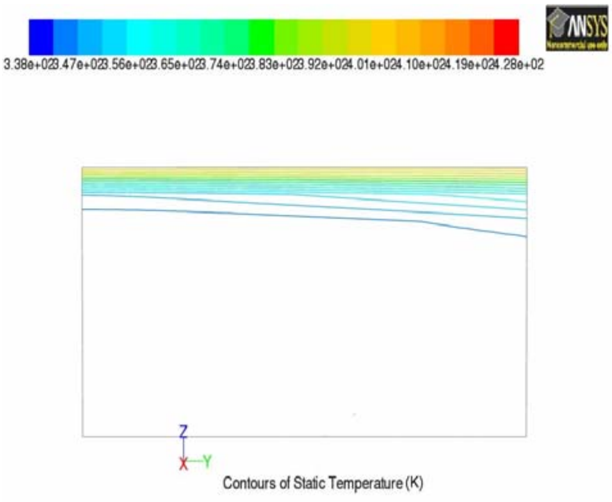

Figure 12: temperature field in a section along the length in the centre of the microchannel

5. Conclusions

A set of experiments were conducted at the Institute of Micro Process Engineering (IMVT) on three self-similar devices, under 5 different conditions in terms of thermal power and 4 different conditions for the mass flow rate.

The results indicate that the definition of the exchange area is not unique and especially in industrial applications, this could lead to misleading information regarding the characteristics of the cooler. However, not the entire area of the device contributes to the heat removal in the same way, and identifying the various contributions will help in defining the correct exchange area. A good approach will probably lead to a redefinition of the heat flux, not in terms of heat exchanging area, but more convincingly in terms of volume, as from an industrial point of view the volume occupied by a device is an intrusive detail.

Another interesting result is the small pressure drop for a device of this size and for the mass flow rate considered. In fact, this is a result mainly due to the quite short length of each channel (0.2mm) reducing the losses due to friction. Conversely, taking the smallest area (of the 2 considered) where the heat exchange can take place was compensated by dramatically increasing the number of microchannels.

6. References

[1] Tuckerman, D.B., Pease, R.F.W., High-performance heat sinking for VLSI, Electron Device Letters, IEEE, vol.2, no.5, pp. 126-129, May 1981

[2] Brandner, J.J., Anurjew, A., Bohn, L., Hansjosten, E., Henning, T., Schygulla, U., Wenka, A., Schubert, K., “Concepts and Realization of Microstructure Heat Exchangers for Enhanced Heat Transfer”, Experimental Thermal and Fluid Science, 30, 2006, 801-809

[3] Kandlikar, S.G., Upadhye, H.R., Extending the heat flux limit with enhanced microchannels in direct single-phase cooling of computer chips, Semiconductor Thermal Measurement and Management Symposium, 2005 IEEE Twenty First Annual IEEE , pp. 8- 15, 15-17 March 2005

[4] Kamaruzaman, N.B., Schygulla, U., Brandner, J.J., Comparison of experimental and calculation results of heat transfer and pressure drop for a microstructure surface cooler”. 11th International Conference on Microreaction Technology (IMRET 11), Kyoto, J, March 8-10, 2010

[5] Wang, Y., Ding, G. F., “Experimental investigation of heat transfer performance for a novel microchannel heat sink Journal of Micromechanics and Microengineering, Volume 18, Number 3, March 2008 , pp. 35021-35028(8)

[6] Xu, J.L., Gan, Y.H., Zhang, D.C., Li, X.H., Microscale heat transfer enhancement using thermal boundary layer redeveloping concept, International Journal of Heat and Mass Transfer, Volume 48, Issue 9, April 2005, Pages 1662-1674

[7] Guo, Z. Y., Li, D. Y., Wang, B. X., A novel concept for convective heat transfer enhancement, International Journal of Heat and Mass Transfer, Volume 41, Issue 14, July 1998, Pages 2221-2225

[8] Shah, R. K., and London, A. L., Laminar Flow Forced Convection in Ducts, 1978, Academic Press, New York, NY.

[9] Squires, T. M., Quake, S. R., Microfluidics: Fluid physics at the nanoliter scale. Reviews of Modern Physics, 2005, 77 (3). pp. 977-1026.