MICROCHANNELS IN MACRO THERMAL MANAGEMENT SOLUTIONS

by

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Modern progress in electronics is associated with increase in computing ability and processing speed, as well as decrease in size. Future applications of electronic devices in aviation, aerospace and high performance consumer products' industry demand on very stringent specifications concerning miniaturization, component density, power density and reliability. Excess heat produces stresses on internal components inside the electronic device, thus creating reliability problems. Thus, a problem of heat generation and its efficient removal arises and it has led to the development of advanced thermal control systems. Present research analyses a thermodynamic feasibility of micro capillary heat pumped networks in thermal management of electronic systems, considers basic technological constrains and design availability, and identifies perspective directions for the further studies.

Computer Fluid Dynamics studies have been performed on the laminar convective heat transfer and pressure drop of working fluid in silicon micro-channels. Surface roughness is simulated via regular, constructal, and stochastic models. Three-dimensional numerical solution shows significant effects of surface roughness in terms of the rough element geometry such as height, size, spacing and the channel height on the velocity and pressure fields.

Key words: *micro-channels, roughness, computer fluid dynamics, constructal principle, micro-capillary heat pumping networks, thermal management*

Introduction

Thermal management is a crucial facilitating technology in the development of modern electronics. It has enabled many of the so-called *Moore's law* advances in computers and microelectronics, which took place in the last years. As the power supplied to electronic packages and microelectromechanical system (MEMS) type devices increases, the problem of thermal control becomes critical. Assembly and packaging technologies are being driven to simultaneously meet very demanding requirements in the areas of performance, power, junction temperature, package geometry, and cost. These demands, plus increased reliability expectations, will push the cooling and packaging limits of electronic products. Conventional methods of heat removal are simply not capable to deal with the thermal gradients, which lead to material failure. It therefore becomes necessary to look beyond traditional thermal management solutions.

From this point of view, liquid-vapor phase change, direct and indirect liquid cooling, impinging jets, droplets, and sprays are attractive cooling options for removing high heat fluxes because of their associated heat transfer coefficients. Liquid cooling with boiling has been extensively studied in the past, starting with the pioneering works of Park and Bergles [1], Carvalho and Bergles [2], Bergles and Bar-Cohen [3], Incropera [4], and Bar-Cohen [5]. The main issues investigated are the critical heat flux levels that can be attained, temperature overshoot and incipient excursion, bubble growth and departure as well as the effect of surface enhancement. Ma *et al.* present an extensive literature survey of jet impingement cooling with and without boiling, [6]. Enhanced surfaces have been investigated at length by Ravigururajan and Bergles [7] and several other researchers, and corresponding surveys have been published by Webb [8]. Enhanced surfaces have proven successful in inducing higher heat transfer rates and lower temperature overshoots in pool and jet impingement boiling. In the context of electronics cooling, Nakayama *et al.* [9] investigated the effect of nucleation sites and designed enhanced surfaces that achieved critical heat fluxes above 100 W/cm^2 with fluorinate dielectric liquids. In this respect Amon *et al.* [10] determined the optimal configuration of micro-structured silicon surfaces, which could be used as the heat spreaders.

Effective indirect liquid cooling with two-phase heat transfer has been realized by means of heat pipes and thermosyphons, [11]. Thus, due to the limited space and power available in laptops, heat pipes are ideally suited for cooling the high power chips. Three or four millimeter diameter micro heat pipes can effectively remove the high flux heat from the processor and spread the heat load over a relatively large area heat sink, where the heat flux is so low that it can be effectively dissipated through the notebook case to the ambient air. Moreover, modern trends within the subsequent wick pumping technology make the perspective of Micro-Capillary Heat Pumped Networks (MCHPN's) for thermal management of electronics both promising and realistic, [12].

Basically, two-phase heat transfer, involving evaporation of a liquid in a hot region and condensation of vapor in a cold region, can provide the removal of much higher heat fluxes than can be achieved through conventional forced air-cooling, which is the reason why considerable research has been redirected towards these approaches for thermal management of electronics. Therefore, in recent years the entire group of devices, based on the use of features and advantages of two-phase heat transfer in capillary-porous structures has been created. Since majority of the devices related to this group has been developed rather recently, their studies basically have constricted experimental background, directed to studying of distinctive principles of heat and mass transfer. This statement represents an essential draw back for execution of the detailed thermodynamic design of such systems and the respective thermodynamic optimization. From this point of view the goal of the present paper is not a raw attempt to suggest a panacea for the resolution of all problems in the application of the MCHPN's at one stroke, but to arrange the respective accents, allowing as to overcome a gap within conventional methods of the thermodynamic analysis of cooling systems, and to create preconditions for its further development. Subsequently, first we shall list some important features of capillary-porous structures, making them as attractive construction element for the present and future thermal management systems.

Capillary-porous structures

Many natural materials fall in the category of porous media, *e. g*., rocks, sands, soils, woods, *etc.* [13]. Many structures made of packed or sintered solid particles, such as sphere packs or fiber mats, and foams are artificial porous media, commonly used in many industrial processes, *e. g*., heat transfer, gas separation, petroleum engineering, *etc*. As a matter of fact, almost all natural and artificial materials are porous, in some sense, 100% perfect solid material is hardly existent. The term *porous solid* is even used to denote crystal structures with molecular-level pores that allow transportation of molecules through those small pores. Hence, particular magnitude of porous media in many industrial processes (*e. g*. heat transfer, catalyst, distillation, *etc*.) is due to the existence of pores.

The pore is a void space, which is not occupied by solid. A quantitative measure for the amount of pore space in a material is porosity, which is defined by a ratio of the void space, V_p, to nominal solid space, $V, \varepsilon \quad V_p / V$. Through these pores, fluids can be transported in vapor or liquid phase. Due to the complexity of pore structure, mass transport occurs through complex path composed of many pores. The existence of pores in a solid facilitates fast heat transfer or chemical reaction by increasing the contact area between solid and fluid. The specific solid/fluid surface area, A_{12}/V is much larger than nominal surface area, and easily exceeds 10^3 m^2/m^3.

Recent progress in thermal sciences allowed prediction and discovering of new advantages of the solid capillary-porous structures associated with their possible application as the construction elements for power generation, refrigeration and heat pumping systems' design [14-16]. Some attractive options for micro-capillary structure application in advanced thermal management solutions are listed in brief below.

Numerous modern fluidic applications in the areas of electronics, medicine and chemistry call for the development of micro-pumps that can efficiently and reliably handle fluids. To meet this need, a number of micro-pumps fabricated with MEMS techniques have been reported in the literature, [17, 18]. Most of the proposed micro-pumps are just micro versions of traditional pumps, while it is possible to move liquids and constituents in liquids without using any moving parts. Non-mechanical pumps can be designed based on thermally driven liquid-vapor phase change in solid micro-capillary structure. To actuate a fluid flow in a micro-channel, pumping mechanisms based on liquid-vapor phase change have been reported by Takagi *et al.* [19] and Ozaki [20].

A passive circulation of the working fluid in vapor cycles by means of prototype piston driven engine was introduced by Kobayashi [21]. Johnson *et al.* [22] presented a prototype thermosyphon Rankine engine. While, in both papers maximum thermal efficiencies of engines were far less than 10% and the flow circulation was gravity driven, as opposed to being driven by capillary pressure. However, the maximum capillary pressure of conventional solid capillary-porous structures is too low (approximately 100 kPa, due to minimum pore diameters in the order of 10^{-5} m) to be of significant concern for driving of generators in heat engines. Modern advances in nanotechnology have made it increasingly practical to create high-porosity materials with median surface pore diameters in the order of a few nanometers, [23, 24]. Theoretically similar structures applied as capillary pumps in vapor cycles with non-mechanical compression can provide capillary pressures in the order of several million Pascal.

Micro-capillary structures are currently used in wicks of conventional, loop heat pipes (LHP's) and capillary pumped loops (CPL's). The main purpose of a wick is to generate the capillary pumping pressure required to transport a working fluid along a two-phase heat transfer loop. The performance of the wick is set by its pore radius and permeability. The capillary pumping head generated is inversely related to the pore size. However, as the pore size decreases, so does the wick's permeability. This causes an increase in resistance to flow. Accordingly, an optimum must be found for a given application.

Width of the wick is its important feature, which must be optimized. The heat transport capability of the heat pipe is raised by increasing the wick width. The overall thermal resistance at the evaporator also depends on the conductivity of the working fluid in the wick. The wick must also be chemically and physically stable in its operating environment over long periods. This would include exposure to operating temperatures and temperatures involved in fabrication. There are several types of wick structures available, including swaged or extruded grooves, screen mesh, cables/fibers, and sintered powder metal. This list of wick structures' types is in order of decreasing permeability and decreasing pore radius.

Grooved wicks have a large pore radius and a high permeability, as a result, the pressure losses are low, but the pumping head is also low. Grooved wicks can transfer high heat loads in a horizontal or gravity aided position, but cannot transfer large loads against gravity. An example is in instances where the heat-input area of the heat pipe is placed physically above the cooled regions of the heat pipe. Screen-mesh wick structures can be made with finer pores and, therefore, offer improved performance over simple-groove wick structures. Since the installation of the screen is an additional step in the fabrication of the heat pipe and because the process can be tedious, screen-mesh wick heat pipes are slightly more expensive than groove-wick heat pipes. Fibrous materials, like ceramics, have also been used widely. They generally have smaller pores. The main disadvantage of ceramic fibers is that, they have little stiffness and usually require a continuous support by a metal mesh. Thus, while the fiber itself may be chemically compatible with the working fluids, the supporting materials may cause problems.

More recently, interest has turned to carbon fibers as a wick material. Carbon fiber filaments have many fine longitudinal grooves on their surface, have high capillary pressures, and are chemically stable. A number of heat pipes that have been successfully constructed using carbon fiber wicks seem to show a greater heat transport capability. Sintered-powder metal wicks are porous metal structures, approximately 50 percent dense, which have small pore radius and relatively low permeability. Since the size of the particles used in forming the sintered structure can be varied, a tailored high-performance wick can be made using this process. Sintered-powder metal-wick heat pipes can be made to work in any orientation, even with the pipe vertical and the heat source at the top. This capability makes the sintered-powder metal-wick structure a very high-performance system, particularly suitable for applications where the use-orientation of the device is uncertain.

Hall [25] suggested improvement of the capillary capability by using an advanced wick structure – graded wick. While uniform wicks are easy to manufacture but do not provide the maximum capillary capability required in many applications, especially for heat pipes that are operating against gravity and long. Because the liquid vapor pressure differential changes continuously from the evaporator to condenser, a graded wick that corresponds to this change is able to provide the maximum capillary capability and the minimum liquid flow resistance.

Thermal design and optimization of micro-capillary heat pumping networks

Various thermal design tools are now available in the designing of electronic components, packages, and systems. Thermal conduction codes are used to model heat flow and temperatures within a package. Computational fluid dynamics (CFD) codes are used to model fluid flow around and through package assemblies along with the associated pressure drop and heat transfer from exposed package surfaces to the fluid stream. In addition, some CFD codes have conjugate capability making it possible to model thermal conduction within the package structure simultaneously with modeling fluid flow and heat transfer in the cooling fluid. However, further improvements are needed to reduce the time consumed in defining the package geometry and structure and to enter related data preparatory to running a model.

Traditional use of flow network analysis in the design of electronics cooling systems has been discussed by Ellison, [26]. Belady *et al.* [27] presented detailed description of generalized flow network modeling methodology for the prediction of flow, pressure, and bulk temperature distributions in arbitrarily complex networks.

Any electronic cooling system can be represented as a network of components such as ducts, heat sinks, screens, filters, passages within card arrays, fans, bends, and tee junctions. Interconnections of these components correspond to the paths followed by the coolant as it passes through the system. While, an essential disadvantage of this methodology consists in lack of ability to predict the details of flow and heat transfer within a component.

Calculating system-wide flow and temperature distributions requires specification of the flow and heat transfer characteristics for the components used in the network model. The pressure loss in a component represents a function of flow rate with the following equation:

$$\Delta p = \frac{f\rho}{2}\left(\frac{Q}{A}\right)^2 \tag{1}$$

Change in the bulk temperature across a component is calculated by specifying the heat dissipated in that component. Further, the average surface temperature of the component is determined from the surface heat transfer coefficient. The following form of the correlations that relate Nusselt number to the Reynolds and Prandtl numbers are used for this purpose:

$$\mathrm{Nu} = C\,\mathrm{Re}^{m}\,\mathrm{Pr}^{n} \tag{2}$$

The calculation of the heat loss/gain in each link in combination with the imposition of energy balance at each junction enables prediction of temperature distribution in the system.

Original approach to the development of electronics' cooling systems was presented by Bejan [28] in the extent of his constructal theory. The author considered a finite-size volume in which heat was generated at every point and which was cooled through a small patch (heat sink) located on its boundary. Assuming availability of finite amount of high conductivity (k_p) material, the optimal distribution of k_p material through the given volume has been determined such that the highest temperature was minimized. The optimized geometry formed by the slow and fast flow regimes unites all the volume-to-point flows. This single principle of geometric optimization and subsequent construction (growth) is repeated toward stepwise larger volume scales. The end result is the optimized architecture (shape and structure) of the composite that connects the sink point to the finite-size volume. If the heat source discharges itself to one point in unsteady mode, then the constructal minimization of volume-to-point resistance is equivalent to the minimization of the time of discharge, or the maximization of the speed of approach to uniformity (system's internal equilibrium).

Thus, the Bejan's principle of evolution is: *For a finite-size system to persist in time (to live), it must evolve in such a way that it provides easier access to the imposed currents that flow through it*, [29]. For instance, this concept allows a deterministic prediction of the optimum structure of electronics' cooling systems. Similarly, improved thermal spreaders and micro heat pipe arrays can be designed at the constructal theory base to extend heat from concentrated chip heat sources to the larger surface area provided by the module cap or heat sink for removal by the external cooling medium. Simulation of the heat and mass transfer processes in the rough micro-channels arrays could be considered as the theoretical basis for the further development and application of the *Constructal theory* in modern technologies [30, 31].

Design of the velocity fields in rough microchannels

Motion of liquid in microchannel could be described by equations of mass and impulse balances in dimensionless forms as:

$$\frac{\partial U_X}{\partial X} + \frac{\partial U_Y}{\partial Y} + \frac{\partial U_Z}{\partial Z} = 0 \tag{3}$$

$$\begin{aligned}
\mathrm{Re}\ U_X \frac{\partial U_X}{\partial X} \quad U_Y \frac{\partial U_X}{\partial Y} \quad U_Z \frac{\partial U_X}{\partial Z} \qquad \frac{\partial P}{\partial X} \quad \frac{\partial^2 U_X}{\partial X^2} \quad \frac{\partial^2 U_X}{\partial Y^2} \quad \frac{\partial^2 U_X}{\partial Z^2} \\
\mathrm{Re}\ U_X \frac{\partial U_Y}{\partial X} \quad U_Y \frac{\partial U_Y}{\partial Y} \quad U_Z \frac{\partial U_Y}{\partial Z} \qquad \frac{\partial P}{\partial Y} \quad \frac{\partial^2 U_Y}{\partial X^2} \quad \frac{\partial^2 U_Y}{\partial Y^2} \quad \frac{\partial^2 U_Y}{\partial Z^2} \\
\mathrm{Re}\ U_X \frac{\partial U_Z}{\partial X} \quad U_Y \frac{\partial U_Z}{\partial Y} \quad U_Z \frac{\partial U_Z}{\partial Z} \qquad \frac{\partial P}{\partial Z} \quad \frac{\partial^2 U_Z}{\partial X^2} \quad \frac{\partial^2 U_Z}{\partial Y^2} \quad \frac{\partial^2 U_Z}{\partial Z^2}
\end{aligned} \tag{4}$$

Here Reynolds number is determined as Re = $U_m H/\nu$.

For estimation of roughness effect, the Poiseuille correlation for the dimensionless pressure drops has been used as:

$$\Delta P \quad \frac{\Delta p H}{\mu U_m} \quad 12\Delta X \tag{5}$$

The calculations were based on the CFD methods (www.ansys.com/cfx) for a 3-D case. Figure 1 presents breaking up of the space region on the finite elements for the constructal (fig. 1a) and regular (fig. 1b) coverages. The calculations have been conducted on the lattice of 32 32 32 cells.

The height of the computable cell's volume from the considerations of symmetry was chosen as equal to the half of the channel height, H. Thus, symmetric boundary conditions where defined as:

$$\begin{aligned}
\frac{\partial U_X}{\partial Z} \quad 0, \ \frac{\partial U_Y}{\partial Z} \quad 0, \ U_Z \quad 0 \ \text{ at } \ Z \quad 0 \\
\frac{\partial U_X}{\partial Z} \quad 0, \ \frac{\partial U_Y}{\partial Z} \quad 0, \ U_Z \quad 0 \ \text{ at } \ Y \quad c/H \\
\frac{\partial U_X}{\partial Y} \quad 0, \ \frac{\partial U_Z}{\partial Y} \quad 0, \ U_Y \quad 0 \ \text{ at } \ Y \quad 0.5
\end{aligned} \tag{6}$$

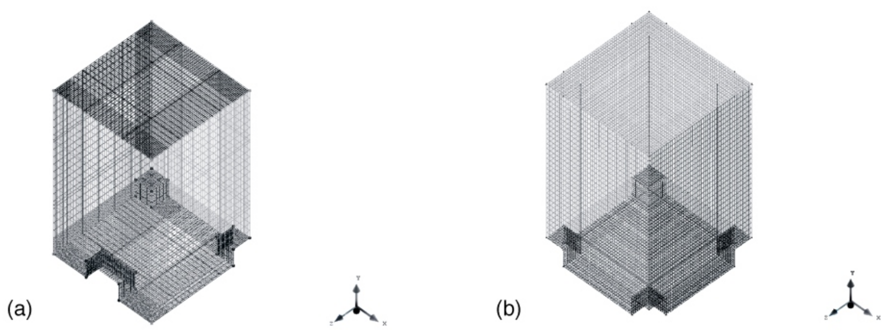

Figure 1. (a) 3-D calculation lattice for constructal coverage; (b) 3-D calculation lattice for regular coverage

The condition of adhesion's absence corresponds to the equality to the zero of all velocity components at $Y = 0$. The periodic boundary conditions introduced in paper [32] are represented by the set of correlations as:

$$\begin{aligned} U_X(X,Y,Z) &= U_X(X+d,Y,Z) \\ U_Y(X,Y,Z) &= U_Y(X+d,Y,Z) \\ U_Z(X,Y,Z) &= U_Z(X+d,Y,Z) \end{aligned} \tag{7}$$

The directions of streams in the element of microchannel with the designed roughness of surface are illustrated in fig. 2. In the beginning, the calculations were conducted for the following conditions:

$$X = 0,\ U_X = 1,\ U_Z = U_Y = 0 \tag{8}$$

$$X = b/H,\ U_X = 1,\ U_Z = U_Y = 0 \tag{9}$$

The derived boundary conditions at the outlet were matched to the inlet conditions in order to afford the periodicity [30]. For the last iteration the following conditions were executed:

$$\begin{gathered} X = 0,\ U_{X|X=0} = U_{X|X=b/H}, \\ U_{Y|X=0} = U_{Y|X=b/H},\ U_{Z|X=0} = U_{Z|X=b/H} \end{gathered} \tag{10}$$

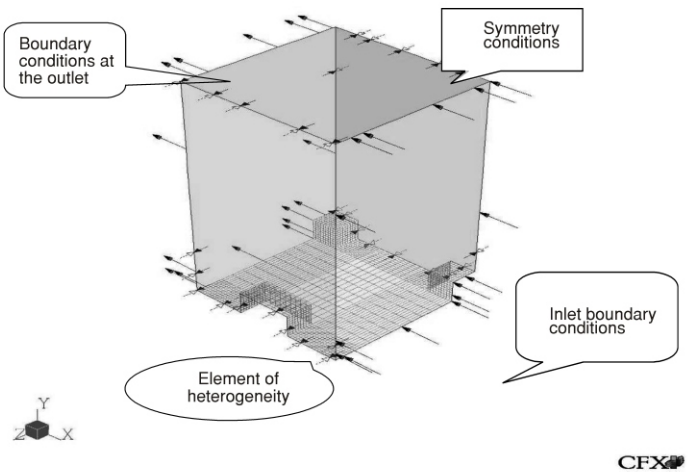

Figure 2. Boundary conditions in the problem of flowlines around of heterogeneous surface

$$X \quad b/H, \quad \frac{\partial U_X}{\partial X} \quad 0,$$
$$U_{Y|X \;\; 0} \quad U_{Y|X \;\; b/H}, \quad U_{Z|X \;\; 0} \quad U_{Z|X \;\; b/H} \qquad (11)$$

Results of calculation of the velocity and pressure fields in micro-channels

For testing of the simulation data, the flow in a smooth channel has been considered and calculation results were compared to the analytical model. Calculations have been executed in the range of the Reynolds numbers of 0.1… 100. For the chosen calculable lattice the minimum error at Re = 0.1 was equal to 0.1%, when the maximum inaccuracy at Re = 100 was equal to 0.3%. Figure 3 presents characteristic distribution of the velocity field for the regular coverage, demonstrating periodic character of the solution.

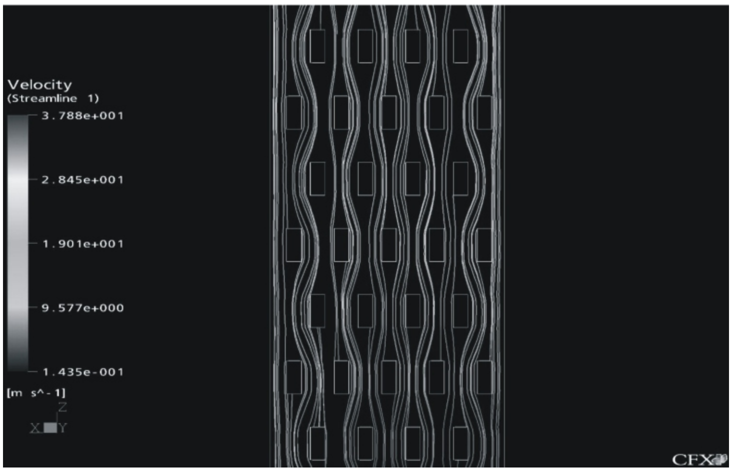

Figure 3. Velocity distribution along the regular heterogeneous surface

Geometrical sizes that more precisely correspond to the descriptions of the real micro-fluidic devices were varied as:

height of the rectangular micro-channel, *H*: 10…100 m,
height of the element of heterogeneity, *h*: 0.1 … 2 m,
width, *a* : 0.1… 1 m, and
distances between the elements of roughness through the length and breadth of stream – (*b* = *c*) were chosen as equal (5 … 50 m).

The 3-D behavior of flowlines around the elements of heterogeneous surfaces is represented in figs. 4 and 5. The stagnation areas in the neighboring of elements of heterogeneity are appeared clearly here.

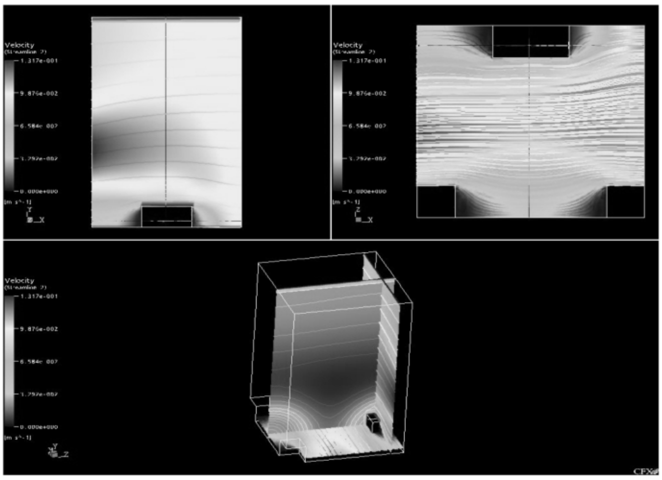

Figure 4. Velocity fields for the constructal model of roughness (H = 5 m, h = 0.5 m, a = 1.0, b = 2.0 m)

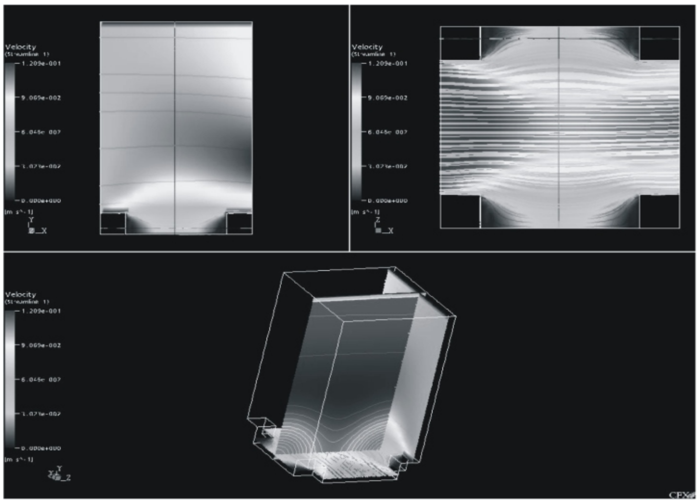

Figure 5. Velocity fields for the regular model of roughness (H = 5 m, h = 0.5 m, a = 2 m, b = 2 mm)

The effect of height of roughness' element is most substantial at the flow of liquid in micro-channels. The results of calculations of pressure gradient depending on the height of roughness and side size of heterogeneity element for the regular model of roughness are represented in fig. 6. For certainty in calculations the value of distance between the elements of heterogeneity was taken as $b = c = 10$ m. It was chosen on the basis of numeral experiments, in order to eliminate the mutual influence of different factors on the change of pressure gradient. At more low values of b, deviations from the Poiseuille flow's mode become evident and must be also taken into account. If the values of b are exceeded 20 m, the correlation between the elements of heterogeneity is unimportant and flow in micro-channels with such roughness is similar to the flows in smooth channels. Sharp growth of pressure drop is observed, when the attitude of height of roughness toward the height of channel exceeds 20%. If the attitude of height of heterogeneous element toward the height of channel, h/H makes 0.5, deviations of the pressure gradient from the Poiseuille flow appear in 20 times higher. On the other hand, the increase of microchannel's height results in diminishment of roughness effect on the flowlines of working liquid. Conventional correlations for smooth channels become fully suitable for the calculations of flows in micro-channels, if the value h/H becomes less than 0.2.

Hence, distinction between the models of heterogeneity to be used appears not so substantial in comparison with the effect of geometrical parameters of heterogeneity. High-quality picture of flow motion for the constructal and casual models of roughness a slightly differs from the regular picture and conduces to more sharp alternation of areas of flow expansion and compression that creates pre-conditions for the severe losses of kinetic energy. Divergences between different models of the surface heterogeneity are not so significant, and for the ranges of geometrical parameters considered in the present research did not exceed 20%. Maximal differences for the resulted gradients were appeared at the ratios of $h/H > 0.4$. Besides, the indicated distinctions do not have systematic char-

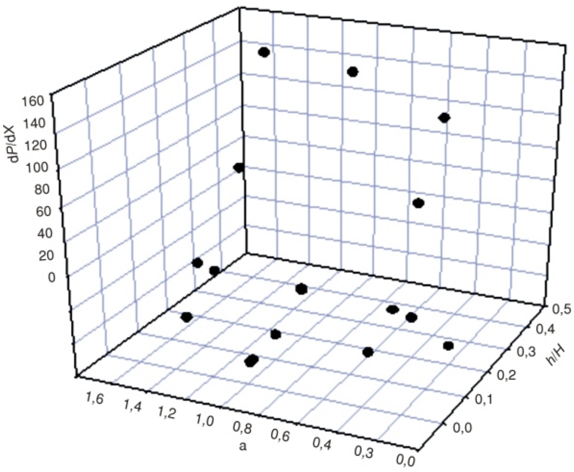

Figure 6. The effect of geometrical parameters of roughness on the dimensionless pressure gradient

acter, and effect of sizes of heterogeneity elements on the pressure gradient can appear even reverse (for example, at the small values of parameter a the value of $\Delta P/\Delta X$ for the regular model of heterogeneity appears higher than for the constructal and casual, while at the large values of a the reverse picture emerges).

The typical pressure distribution in the microchannel is presented in fig. 7, where zones of high and low pressures are visualized. The presence of such zones conduces to distortion of flows in the micro-channels and reflects the significant role of roughness, influencing the temperature fields.

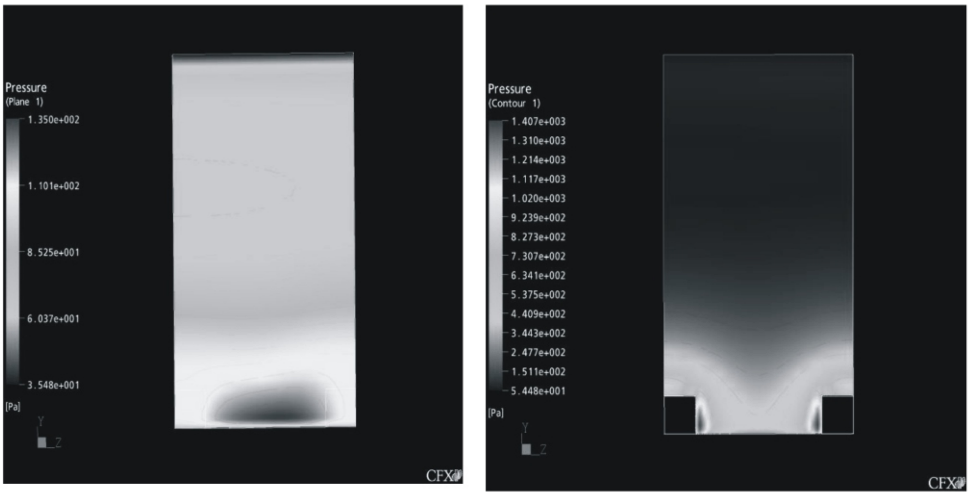

Figure 7. Pressure distribution in micro-channel

Micro-capillary heat pumped networks technology

Thermosyphon collectors (TC's) and collector heat pipes could be considered as the first-level approximation to the construction of MCHPN's. TC's have been designed in order to achieve a uniformity of the temperature distribution within the electronic assemblies and packages [33, 34]. Basic principle of the mentioned designs is integration of internal volumes of separate heat pipes or thermosyphons into the mutual volume, which forms a joint heat removal zone. Simultaneously, both geometry and internal two-phase heat transfer at each level of the integration design ensure the basic technological regulations. In that order, the TC's provide effective thermal connection and have been core construction elements for positioning of the electronic components. Typical engineering designs of TC's could be classified as follows:

(1) TC's of the assembly-level design designated as primary, fig. 8a, and
(2) TC's of the package-level design designated as secondary, fig. 8b.

Collecting swelling of each TC represents a condenser from where heat is transferred either to other TC's or to the heat sinks, or to the surroundings. A quality of the thermal junction between the surfaces of heat generating elements and TC's represents an extremely important feature. Executed experiments shown that considered TC's designs reduce the volume of electronic devices and temperature overshoots up to 1.5…2 times, achieving the high uniformity of temperature fields as for the primary TC's both the secondary one, *i. e.* improve the reliability of electronics significantly.

Micro heat pipe array application as integrated part in the cooling of microelectronics is well known and could be considered as the second-level approximation to construction of MCHPN's. The micro heat pipes built directly within the substrate material transfer heat away from the chip much more efficiently than conduction through solid copper or transport through larger heat pipes on the substrates surface and thus allow for greater heat reduction and improved thermal management solutions. The closer the coolant is to the circuit, the more heat can be managed. As the coolant directly underneath the operating micro chip heats up and evaporates, it moves through the paths to cooler areas, where it recondenses, distributing its heat throughout the substrate. The amount of capil-

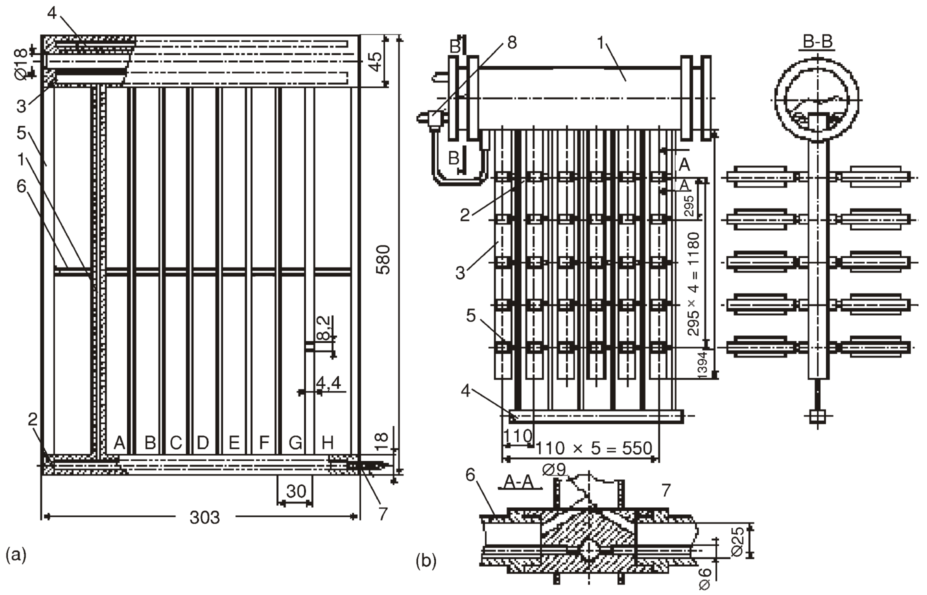

Figure 8. Thermosyphon collectors

(a) Primary thermosyphon collector 1 evaporative channels (A, B, C, D, E, F, G, H); 2 bottom collector; 3 top collector; 4 condenser; 5, 6 edges; 7 charging tube; (b) Secondary thermosyphon collector: 1 tube condenser; 2 hauled down tube; 3 lifted tube; 4 bottom collector; 5, 6 evaporative channels; 7 junction of the evaporative channels with tubes 2 and 3; 8 regulator of condensate level; 9 plates for arrangement of electronic components

lary pressure inside the tubes is a function of the size of the passages. As the structures get smaller, more pumping action is created within the substrate. A coolant and micro pipe geometry can be selected that best transfers heat given each device's design and operating temperature range.

Mallik *et al.* [35] developed a transient three-dimensional finite difference model to evaluate the percent reduction in the maximum chip surface temperature, the mean surface temperature, and the thermal gradient occurring on the chip for 100 μm diameter pipes. They observed significant reductions in the maximum chip temperature of up to 40 percent and in the transient response time, pointing up the improvement of the heat removing capability. Peterson *et al.* [36] performed experiments on micro heat pipe array. Their results proved clearly that an array of micro heat pipes as an integral part of silicon wafers can significantly improve the performance and reliability of semiconductor devices, by increasing the effective thermal conductivity, decreasing thermal gradients, and reducing the intensity and number of localized hot spots.

As it was mentioned, further improvements in thermal design and optimization of micro heat pipe arrays for cooling of electronics should be accomplished by means of the constructal principle of shape and structure formation. Keeping in mind the constructal principle and wick pumping technology tools, a next step to MCHPN's design could be developed by using the novel solutions from microengineering and nanotechnology (see fig. 9).

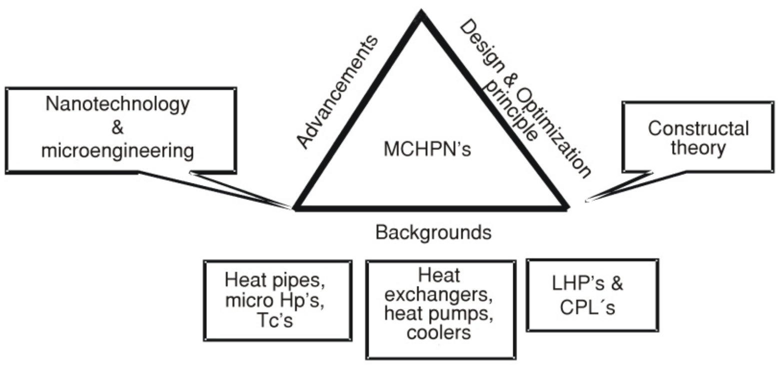

Figure 9. New generation of micro-capillary heat pumping networks

Concluding remarks

Conventional cooling technologies in the electronic industry have limitations on removing non-uniform, high heat fluxes from the surface of microprocessors. The combination of high heat fluxes with the non-uniformity of heat dissipation requires technologies able to remove large amounts of heat in a spatially and temporally variable manner.

Thermal management system for advanced electronics needs to meet the selection criterion of reliability, simplicity, low cost, and above all, effectiveness. Accounting a huge number of micro components with vary heat generation intensities and contradictory temperature regimes within a complex electronic system, thermodynamically proved, optimum thermal management system appears a complex network of cooling, heat dissipating, transfer and pumping elements. Different ideas have been used to develop such high performance thermal management systems. Researchers try to utilize the passive forces in the liquid such as capillary effect, osmotic effect, viscosity effect, and expansion effect to create self circulating cooling and heat dissipating devices.

Since Cotter [37] proposed using micro heat pipes for heat removal in semiconductor materials, a number of steady state and transient models for micro heat pipes have been proposed, tested and verified. Micro heat pipes have potential to solve the overheating problem at submicron level. Further improvements in thermal design and optimization of micro heat pipe arrays for cooling of electronics should be accomplished by means of the constructal principle of shape and structure formation.

Velocity, pressure and temperature fields occurring in micro- and macro-channels at the analysis of different models of heterogeneous surfaces (regular, casual, and constructal) confirm the similarity of impulse transfer processes at identical boundary conditions. Thus, conventional correlations for the heat transfer coefficients and hydraulic resistances at the flows in channels remain applicable for micro- and macro-channels case, if boundary conditions and geometrical limitations are properly formulated. Existing divergences in experimental data of different authors are connected with difficulties in reproduction of identical experimental conditions and their accuracy, instead of new physical phenomena that could be related to the micro-scales, and the recent experimental data just confirmed such standpoint [38].

It is clear that the following progress in microelectronics should be associated with the integration of micro-cooling, heat pumping, micro-channels and other micro-devices into the entire electronic system. Naturally, engineering thermodynamics' state of the art with respect to the design and optimization of cooling, refrigeration and heat pumping systems both with mechanical and non-mechanical compression principles will represent a significant knowledge-base for the novel MCHPN's developments. Probably the next generation of the MCHPN's will represent a complete analogy to abiotic skin for micro-electronic device. Thus, using of capillary-porous structures as the construction element and two-phase liquid-vapor transition as the essential heat transfer mechanism will perform a realistic opportunity for the effective network integration in the nearest decades.

Nomenclature

A – flow area , [m^2]

A_{12} solid/fluid surface area, [m^2]

a	width of the channel, [m]
b, c	distances between the elements of roughness through the length and breadth of stream, [m]
C, m, n	constants, [–]
f	loss coefficient, [–]
H	height of the channel, [m]
h	height of the element of heterogeneity, [m]
k_p	thermal conductivity, [$Wm^{-1}K^{-1}$)]
Nu	Nusselt number, [–]
p	pressure drop, [Pa]
Q	volumetric flow rate, [m^3/s]
Pr	Prandtl number, [–]
Re	Reynolds number, [–]
U_m	average velocity of liquid flow in the minimum cross section of the channel, [m/s]
U_X, U_Y, U_Z	dimensionless components of flow velocity along side the coordinate axes, rated to U_m, [–]
V	nominal solid space, [m^3]
V_p	void space, [m^3]
X, Y, Z	dimensionless coordinates, rated to H, [–]

Greek symbols

ε	porosity, [–]
ν	kinematic viscosity, [m^2/s]
μ	dynamic viscosity, [Ns/m^2]
ρ	fluid density, [kg/m^3]

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