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RELATING SEMICONDUCTOR HEAT SINK LOCAL AND NON-LOCAL EXPERIMENTAL AND SIMULATION DATA TO UPPER SCALE DESIGN GOALS

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Abstract

The primary difficulty in semiconductor heat sink (and many other types of heat exchangers) research and design is not a lack of interest or money, but rather confusion with what being looked for and adequacy of the tools used for the search. As recently shown, there are few meaningful parameters (apart from sizes and weight) or physical characteristics of interest in semiconductor cooler design are local values. Even the maximum temperature of the base T_{max} or semiconductor temperature are not local. In this work outlined the description in detail of arguments on how, and for what reasons, the measured data are to be simulated or measured and represented in a way that allows design goals to be formulated primarily with bulk physical characteristics. We demonstrate why studies of only averaged local integrated variables are not enough. Four sample semiconductor heat sinks of two morphologies (three samples of round pin fin and one sample of longitudinal rib fin sinks) were studied by different techniques and models. There were changes in by-pass values, external heat flux and flow rate. The results are depicted with using new parameters that better represent the needs of a design process as well as the usual parameters used in the past. Characteristics reported are the heat transfer rate in solid phase, relative fin effectiveness, and influence of only morphology features among others. Some suggestions for heat sink design are discussed.

Nomenclature

c_p	-	specific heat $[J/(kg \cdot K)]$
d_h	-	hydraulic diameter $[m]$
d_{por}	-	$= 4 < m > /S_w$ characteristic length $[m]$
dS	-	interphase differential area in porous medium $[m^2]$
∂S_w	-	internal surface in the REV $[m^2]$
$\widetilde{f}$	-	averaged over $\Delta\Omega_f$ value f
$< f >_f$	-	value f, averaged over $\Delta\Omega_f$ in a REV
$\widehat{f}$	-	value f morpho-fluctuation in a Ω_f
f_f	-	Fanning friction factor of momentum resistance in the volume [-]
E_{eff1}	-	effectiveness of heat transfer per unit volume $[1/K]$
H_r	-	heat transfer rate per unit volume per unit temperature difference $[W/(m^3K)]$
K_m	-	turbulent eddy viscosity $[m^2/s]$
K_T	-	turbulent eddy thermal conductivity $[W/(mK)]$
$\langle m \rangle$	-	averaged porosity [-]

Nu_w	-	bottom wall Nusselt number [-]
P_p	-	pumping power per unit of volume $[W/m^3]$
q_w	-	heat flux through the bottom surface of the heat sink $[W/m^2]$
p	-	pressure $[Pa]$
Re_{ch}	-	Reynolds number of pore hydraulic diameter [-]
S_{all}	-	internal surface plus bottom wall wetted surface $[m^2]$
S_w	-	specific surface of a porous medium $\partial S_w/\Delta\Omega$ $[1/m]$
S_{wb}	-	bottom wetted surface [m2]
S_{wint}	-	internal surface [m2]
S_w^*	-	overall specific surface per unit volume of heat exchanger [1/m]
T_{in}	-	inlet temperature of the coolant [K]
$T_{\max}$	-	maximum temperature of the wall [K]
$\widetilde{\overline{U}}_b$	-	averaged interstitial velocity [m/s]

Subscripts

f	-	fluid phase
i	-	component of turbulent vector variable
L	-	laminar, lower scale
m	-	scale for nondimensionalization
s	-	solid phase
T	-	turbulent
w	-	wall

Superscripts

$\sim$	-	value in fluid phase averaged over the REV
$-$	-	mean turbulent quantity
u	-	upper scale
$*$	-	nondimensional value

Greek letters

α_T	-	turbulent heat transfer coefficient $[W/m^2K]$
$\overline{\alpha}_{int}$	-	mean heat transfer coefficient in volume of heat sink (averaged over the internal surface) $[W/m^2K]$
α_w^*	-	bottom surface (wetted plus through the solid phase) heat transfer coefficient $[W/m^2K]$
Ω	-	volume of the heat sink $[m3]$
$\Delta\Omega$	-	representative elementary volume (REV) $[m^3]$
$\Delta\Omega_f$	-	pore volume in a REV $[m_3]$
ν	-	kinematic viscosity $[m^2/s]$
ν_T	-	averaged turbulent eddy viscosity $[m^2/s]$
ρ_f	-	density of the coolant fluid$[kg/m^3]$

Introduction

The primary goal in semiconductor heat sink design is simple. It is to increase the heat transfer while decreasing the momentum resistance as for regular closed type heat exchangers is the goal. Nevertheless, as soon as everyone agrees that the best way to achieve the maximum heat transfer rate within a particular volume of heat sink is through the introduction of additional heat exchanging elements (ribs or pins of different shape) the problem becomes a two scale heterogeneous volumetric heat exchanger design problem. The processes on the lower scale heat transport - which is the local convective and conductive heat transfer inside and around transfer elements (ribs, fins), are no longer describe the heat transfer rate of the whole sink. We name here the upper scale of the device the spacial scale which is usually corresponds to one of the device' sizes. The semiconductor heat sink has three sizes of the scale ˜0(10cm), also the properties of the device performance as heat transfer rate or efficiency are related to or considered at as in the upper scale. At the same time, the formulation of the problem of a heat sink for a one-temperature, or even a two-temperature homogeneous medium does not involve or connect the local (lower scale) transport characteristics determined by the morphology of the surface elements, directly to the performance of heat sink nor does it give guidance on how to improve the performance characteristics.

In our effort to tie the experimental characteristics of heat sink to the theoretical scaled (VAT) description and simulation of semiconductor base-to-air heat sinks, we came to the process of coupling of two scale modeling and experiment for heat sink design. Most past work focused on the upper scale performance characteristics resulting in many efforts to measure the bulk heat transport rate and in modeling of numerous morphologies (see, for example, Andrews and Fletcher (1996), Bejan and Morega, (1993); Bejan, (1995); Fabbri (1999); Jubran, Hamdan, and Abdualh (1993); Kim and Kim (1999); You and Chang (1997), etc.). In the two scale set up, VAT upper scale governing equations applicable to this problem, contain four additional descriptive terms in the momentum equation (for 1D turbulent equation), seven terms in the fluid temperature equation, and five additional terms in the solid phase (reflecting heat transport through ribs, pins) temperature equations (Gratton et al. 1996; Travkin and Catton, 1999; Travkin et al., 2000).

At the present time, only few first experimental setups needed for development of experimental technique for VAT heat exchanger study were performed (Rizzi et al., 2001; Travkin et al., 2001b). Contrary to simulation numerical experiments, the physical experiment is usually much more restrictive in terms of the number of local experimental points that can be obtained. It is a problem to properly make local measurements and to relate the measurements within the volume of the heat exchange device to the results from simulations because the data point is a pint value and the simulation value is an average over a volume of finite size. In this modeling effort and experiment we attempt to deal with both using the two-scale approach.

We analyze effectiveness models by Andrews and Fletcher (1996), parameters by You and Chang (1997), Fabbri (1999), among others, in effort to reveal the positive features in them. Andrews and Fletcher (1996), provide comparisons of a wide variety of heat enhancing technologies based on the parameter of heat transfer rate per unit volume per unit temperature difference ($S_{all}\overline{\alpha}_{all}/\Omega$) and pumping power per unit volume P_p/Ω .

The two morphologies we dealt in this study of heat sink design were researched numerous times based on conventional one scale heat transfer-fluid mechanics descriptions see, for example, among others the works by Bejan and Sciubba (1992), Bejan and Morega (1993), Bejan (1995, 1999), Kim and Kim (1999), You and Chang (1997). The major problem with this approaches is that their formulations (and consecutive experimental or theoretical studies) are done on the lower scale (homogeneous) of description - but the answers have been sought for the upper scale - general scale of the heat transfer device. As we mentioned, this gives the gap between the formulations and the goals.

In our previous studies (Travkin et al., 2000, 2001a,b) and in the current one we do not need to compare any of the local characteristics or functions (although we obtained all of the functions in their corresponding models). We are mostly interested in question of how the device behaves in experiments and in the corresponding mathematical simulation as a whole unit. At the same time we are not engaged into the balance studies conventional in the heat exchangers technology.

Heterogeneous VAT Based One Phase Analysis of Semiconductor Heat Sinks Experimental Data for Two Morphologies of Semiconductor Heat Sinks

Using the heterogeneous media simplified VAT performance models and characteristics for heat transfer in a flat channel with non specified medium morphologies of heat transfer volumetric results in the following:

a) overall heat transfer rate through the fluid phase per unit volume per unit temperature difference in the device

$$H_r = \frac{S_{all}\overline{\alpha}_{all}}{\Omega}, \quad \left[\frac{W}{m^3K}\right], \tag{1}$$

where S_{all} is the total heat exchanging internal (including the bottom wetted surface) surface, $\overline{\alpha}_{all}$ is the combined (averaged over the all internal and bottom surfaces) heat transfer coefficient, and Ω is the volume of heat exchanger; and

b) pumping power per unit volume of the sink $P_p = P/\Omega,\ [W/m^3]$;

c) effectiveness parameter for heat transfer through the fluid phase for the volume of the heat sink is

$$E_{eff1} = \frac{H_r}{P_p}, \quad \left[\frac{1}{K}\right].$$

As such the characteristics' formulae given above are the same as used for heat exchangers performance elsewhere (see, for example, Andrews and Fletcher, 1996). Given that we use the two scale VAT approach their detailed mathematical formulations would be very different as disclosed below.

Important characteristic to evaluate in the heat exchange device, is the heat transfer rate H_r for a known heat flux q_w through the bottom surface of the heat sink

$$H_r = \frac{S_{all}\alpha_w^*}{\Omega} = S_w^* \frac{q_w}{(T_{wmax} - T_{in})} = Nu_w \frac{k_f}{d_{por}} S_w^* \sim$$
$$\sim Nu_w \frac{k_f S_w}{4\langle m \rangle} S_w^*, \quad \left[\frac{W}{m^3 K}\right], \tag{2}$$

where the value of wall heat transfer coefficient α_w^* $\left[\frac{W}{m^2K}\right]$ (artificial actually heat transfer coefficient) we do not need to know because we use the value of assigned bottom wall heat flux q_w $\left[\frac{W}{m^2}\right]$ which is assumed as known. We determine it through the electrical power input. We recognize in this experiment set-up that the proper defined heat transfer coefficient α_{all}^* is hardly achievable using current equipment and techniques. At the same time, we are in the position to know exactly the amount of heat transferring to the upper base surface and plate fins volume. In the above expression for H_r still used the overall heat transfer surface S_{all} just to preserve the general homogeneous formulation (1) for the heat transfer rate H_r.

The next interesting feature of this approach is that one can assess the value of Nu_w and consequently the value of α_w^* (wall heat transfer coefficient - which is unknown generally before experiment) as

$$S_w^* \frac{q_w}{(T_{wmax} - T_{in})} = Nu_w \frac{k_f}{d_{por}} S_w^* \Rightarrow$$

$$Nu_w = \frac{q_w d_{por}}{(T_{wmax} - T_{in})\, k_f}, \quad \alpha_w^* = \frac{q_w}{(T_{wmax} - T_{in})}. \tag{3}$$

In this expression for Nu_w one can calculate or knows any variable on the right side. Note - that this is not an internal porous medium heat transfer Nusselt number. It is the bottom wall averaged across both phases Nu_w number. Implicitly it means still - the overall (bottom and internal surface) Nusselt number. Some more detail on evaluation of H_r, P_p, and effectiveness E_{eff1} based on experiment see in Travkin et al. (2001b) and Rizzi et al. (2001).

The heat transfer rate H_r and effectiveness number E_{eff1} has been explicitly used for comparison of our four sink samples - three samples with staggered pin fins and one sample with longitudinal fins - Figs. 1-2.

Under all the test conditions employed, more than 98% of the heat generated in the copper block passed, through the heat sinks, to the air in the wind tunnel duct. To apply the corresponding VAT simulation techniques, temperatures along the pin fins were taken. For each of three pin fins of the pin fin heat sinks along the flow direction, temperatures forward and backward were measured. Furthermore, the same pin fins were drilled to allow the collocation of two wires in order to measure the pin fin temperature at 1/3 and 2/3 of its height. The same technique was used for longitudinal fins sink - Figs. 1-2.

The heat dissipating enhanced surfaces of pin fin samples are made of aluminum with a conductivity of 225 [W/m K], while the longitudinal fin sink has aluminum conductivity 204 [W/m K]. Each of the three pin fin heat sinks had constant fin height 0.0381m, constant fin diameter 0.00318m, but the pitch was varied. All the three pin fin heat sinks tested had a staggered pin fin layout. The procedure was repeated for input powers of 50, 125 and 222 W.

For more detail on experiment set up, measurement techniques, and some preliminary results of data reduction see Rizzi et al. (2001) and Travkin et al. (2001b). Data reduction analysis shown that there are still insufficiencies in the applied above bulk simple performance characteristics and some of them would be improved further.

Heterogeneous Two-Scale Two-Phase VAT Analysis of Semiconductor Heat Sinks Experimental Data

The heat transfer process in a semiconductor heat sink embraces processes in both phases - air and solid phase. Nevertheless, in the only obtained above criteria all belongs and take into account the characteristics of fluid (convective) heat transfer, and not even one coefficient used explicitly which describes the solid phase characteristics. We introduce the heat transfer rate in the solid phase of heat sink. It is based on the same idea of intensity of heat transfer through the solid phase. It is assessed as the heat transfer rate (via the solid phase) per unit volume per unit temperature difference

$$H_{rs} = \frac{S_{ws}q_s}{\Omega\left(T_{wmax} - T_{in}\right)} = \frac{S_{wint}\overline{\alpha}_{int}}{\Omega}$$

$$= \frac{S_{ws}\left(-k_s\frac{\partial T_s}{\partial z}\right)}{\Omega\left(T_{wmax} - T_{in}\right)} == Bi\left(\frac{k_s}{d_{por}}\frac{S_{wint}}{\Omega}\right),$$

$$[\frac{W}{m^3K}],\ Bi = \frac{\overline{\alpha}_{int}d_{por}}{k_s},\ \overline{\alpha}_{int} = \frac{S_{ws}}{S_{wint}}\frac{q_s}{\Delta T}, \quad (5)$$

where S_{ws} is the bottom solid phase (pin fins or ribs etc.) cross-section area which can characterize the heat transfer capability of the internal surface of the heat sink to some extend. Because the internal heat flux can be determined with the two temperature differences we use both as

$$\begin{aligned} q_{int} &= \overline{\alpha}_{int}\Delta T = \alpha_{int}\Delta T_{int}, \\ S_{wint}q_{int} &= \left(\alpha_{int}\Delta T_{int}\right)S_{wint} = \\ &= \left(\overline{\alpha}_{int}\Delta T\right)S_{wint},\ [W], \end{aligned} \quad (6)$$

meaning $\alpha_{int} \neq \overline{\alpha}_{int}$, but we agree with this, because our goal is in no means to find the correct α_{int} - we are doing the different task. In our assessment by Travkin et al. (2001b) we accepted point of view when $S_{wint} = S_{ws}$ which simplified the analysis. Now we want to apply also the following definitions

$$\begin{aligned} \overline{\alpha}_{int} &= \frac{q_{int}}{\left(\Delta T = \left(T_{wmax} - T_{in}\right)\right)}, \\ \alpha_{int} &= \frac{q_{int}}{\left(\Delta T_{int} = \left(T_{int} - T_{in}\right)\right)}, \\ S_{wint}\frac{q_{int}}{\Delta T} &= S_{wint}\overline{\alpha}_{int} = S_{ws}\frac{q_s}{\Delta T}, \\ S_{wint}q_{int} &= S_{ws}q_s, \end{aligned} \quad (7)$$

and

$$q_s = -k_s\frac{\partial T_s}{\partial n} = -k_s\frac{\partial T_s}{\partial z},\ [\frac{W}{m^2}],\ q_s \neq \langle s\rangle q_w,$$

(here T_s meaning the fins medium solid phase temperature $T_s \cong \{T\}_s$).

If we would know q_s then H_{rs} can be evaluated. All other variables are known or experimentally assessable. The value of q_s was assessed through our measurements, having the measurements already in the few sets of two consecutive points along of the pins with known distance between them Δz.

For the pin fin sinks

$$H_{rs} = \frac{S_{ws}q_s}{\Omega\left(T_{wmax} - T_{in}\right)} = \quad (9)$$

$$= \left(-k_s\frac{\partial T_s}{\partial z}\right)\frac{\left(\pi R_{pin}^2\right)n_{pins}}{L_xL_yL_z\left(T_{wmax} - T_{in}\right)}, \quad (10)$$

while for the longitudinal fin sink

$$H_{rs} = \left(-k_s\frac{\partial T_s}{\partial z}\right)\frac{\left(w_{fin}L_x^u\right)n_{fins}}{L_x^uL_y^uL_z^u\left(T_{wmax} - T_{in}\right)}, \quad (11)$$

with fins width $w_{fin} = 6.35mm$, and number of fins $n_{fins} = 8$.

There are the phase based definitions of the effectiveness parameters of solid phase $E_{effs} = H_{rs}/P_p$. For longitudinal fin sink it is

$$E_{effs} = \frac{H_{rs}}{P_p} = \left[Bi\left(32\frac{S_{wint}}{\Omega\langle m_{yz}\rangle}\frac{\langle m\rangle^3}{S_w^3}\frac{k_s\rho_f^2}{\mu^3}\right)\right] \quad (12)$$

$$/\left[f_fRe_{por}^3\right],\ \left[\frac{1}{K}\right], \quad (13)$$

$$\begin{aligned} Bi &= \frac{\overline{\alpha}_{int}d_{por}}{k_s} = \frac{S_{ws}q_sd_{por}}{S_{wint}\left(T_{wmax} - T_{in}\right)k_s} \\ &= \frac{\left(-k_s\frac{\partial T_s}{\partial z}\right)d_{por}S_{ws}}{\left(T_{wmax} - T_{in}\right)k_sS_{wint}}, \end{aligned} \quad (14)$$

which also has the factor $\left(32\frac{S_{wint}}{\Omega\langle m_{yz}\rangle}\frac{\langle m\rangle^3}{S_w^3}\frac{k_s\rho_f^2}{\mu^3}\right)$ which combines the influence of morphology and physical characteristics of both phases.

The bottom flat surface fluid phase heat transfer effectiveness is

$$E_{eff,bf} = \frac{H_{rbf}}{P_p},\ H_{rbf} = \frac{S_{wb}}{\Omega}\frac{q_w}{(T_{wmax} - T_{in})}. \qquad (15)$$

All these parameters like H_r, H_{rs}, H_{rbf} (Figs. 3-4) are directly involved in the VAT mathematical governing equations modeling in a simple way. They are also helpful in delivering some more details in terms of energy balance verification which can not be observed with the only fluid phase heat transfer rate H_r. The amount of heat, for example, dissipated through the fins P_{fins} and the complimentary amount of heat P_{bf} which went to air through the bottom plate of sink #R were compounding to the balances which were not perfectly matched to the input power - Figs. 3-4. These results justified the more advanced implementation of VAT approach.

Heterogeneous VAT Approach to Definition of Criteria for Heat Sink Optimization Problem as in Heterogeneous or Like Medium

As it was discussed in the previous paragraphs the all matter of formulation of performance parameters or optimization parameters was related to the formulation of the mathematical model for the process.

Contrary to the homogeneous medium transport mathematical formulations - the scaled heterogeneous medium mathematical models and governing equations composed in such a way that they contain the additional terms reflecting physical phenomena which are important in heterogeneous medium transport and can not be seen or included in the traditional homogenous formulations. That means, that these additional effects should find their ways to the description or models which reflect the characteristics of performance of the device.

Let's start with the analysis of the variables which are the most looked after in this problem. As one observes that most of variables which usually modeled or measured are non-local as is. Below given few of the most pertinent variables and their actual meanings

$$\begin{aligned} T_s &\equiv \{T_s\}_s,\ \overline{T}_f \equiv \{\overline{T}_f\}_f, \\ \mathbf{q}_{wf} &\equiv \{\mathbf{q}_w\}_f,\ \mathbf{q}_{ws} \equiv \{\mathbf{q}_w\}_s, \\ \overline{\mathbf{V}} &\equiv \{\overline{\mathbf{V}}\}_f,\ H_r \equiv \{H_r\}_{\partial S}, \end{aligned} \qquad (16)$$

$$\begin{aligned} P_p &\equiv \{P_p\}_f = \frac{S_{yz}\{\Delta p\}_f}{\Omega}\{\overline{U}\}_f = \\ &= \frac{S_{yz}\{\Delta p\}_f}{\Omega}\frac{1}{\Delta\Omega_f}\int_{\Delta\Omega_f}\overline{U}\left(\vec{x}\right)d\omega, \end{aligned} \qquad (17)$$

$$\begin{aligned} f_f &= \frac{d_h}{2\,\varrho_f \tilde{\overline{U}}^2}\left(\frac{\Delta P}{L}\right) = \\ &= \frac{2\langle m\rangle}{\varrho_f \tilde{\overline{U}}^2 S_w}\left(\frac{\Delta P}{L}\right),\ d_h \cong d_{por} = \frac{4\langle m\rangle}{S_w}, \end{aligned} \qquad (18)$$

where the phase averaging $\{\mathbf{q}_w\}_f$ and $\{\mathbf{q}_w\}_s$ is implicitly used, while one needs to recognize that the written notations do not reflect correctly the conditions of variable's consideration, calculation or measurement.

Now one can apply some VAT formulae to obtain the real heterogeneous fluid phase heat transfer rate which includes heat fluxes via the interface ∂S_w and via the bottom wetted surface

$$\widetilde{H}_r\left[\frac{W}{m^3 K}\right] = \frac{S_{all}\alpha^*_{all}}{\Omega} =$$

$$\frac{1}{\Omega\Delta T}\left[\int_{\partial S_w}(-K_T)\frac{\partial\overline{T}_f}{\partial x_i}\cdot\vec{ds} - S_{wb}* \right. \qquad (19)$$

$$\left(\widetilde{K}_T\nabla\tilde{\overline{T}}_f + \left\{\widehat{K}_T\nabla\hat{\overline{T}}_f\right\}_f + \frac{\widetilde{K}_T}{\Delta\Omega_{fb}}\int_{\partial S_{wb}}\hat{\overline{T}}_f\vec{ds}\right),$$

this needs to be compared to H_r developed above and used as heterogeneous parameter for unknown bottom heat flux

$$H_r = \frac{S_{all}\alpha^*_{all}}{\Omega} = S^*_w \alpha^*_{all} = = Nu_{all}\frac{k_f S_w}{4\langle m\rangle} S^*_w, \; Nu_{all} = \frac{\alpha^*_{all} d_{por}}{k_f}.$$

Next is the heterogeneous solid phase heat transfer rate H_{rs} assessed as

$$\widetilde{H}_{rs}\left[\frac{W}{m^3K}\right] = \frac{S_{ws}\{\mathbf{q}_w\}_s}{\Omega\left(T_{wmax}-T_{in}\right)} =$$

$$= \frac{S_{ws}}{\Omega\left(T_{wmax}-T_{in}\right)}\left(-k_s\nabla\widetilde{T}_s - \frac{k_s}{\Delta\Omega_{fs}}\int\limits_{\partial S_{wb}} \widehat{T}_s d\vec{s}_1\right). \quad (20)$$

We take here $\{\mathbf{q}_w\}_s$ but not $\langle \mathbf{q}_w\rangle_s = \langle s\rangle\{\mathbf{q}_w\}_s$ because we need the heat flux through the only solid phase, not through the unit of surface. Still the value $S_{ws}\{\mathbf{q}_w\}_s/\Omega$ can be close to the value $\langle \mathbf{q}_w\rangle_s = \langle s\rangle\{\mathbf{q}_w\}_s$.

Comparing to the homogeneous H_{rs}

$$H_{rs} = Bi\left(\frac{k_s}{d_{por}}\frac{S_{wint}}{\Omega}\right), \; [\frac{W}{m^3K}],$$

one would see the difference, for example, in the way the heat flux is taken in both cases. As we can suspect based on out prior VAT simulation (for example, Travkin and Catton, 1998; Travkin et al., 2000b; etc.) that the more correct mathematical VAT formulation would be brought for the simulation the better result one should expect.

Direct Simulation using the VAT 2D Models

Explaining further the variables and definitions used in this two-scale two-phase heterogeneous experimental data reduction procedures for the semiconductor heat sink experimental evaluation we would like to return to published in 1992, 1995 works (Travkin and Catton, 1992, 1995; Gratton et al., 1996) where the model for analogous problem of turbulent heat transfer in a flat channel filled with porous medium was used. There are eight nondimensional Medium Specific Control Functions (MSCF) or parameters for turbulent regime (Travkin et al., 2000) on the upper scale of simulation.

The Simplest Case with Constant $\langle m\rangle$, S_w. Let's write those parameters in a simplified possible way (even using short-cuts obtained in the turbulent statement treatments)

variable	definition	No.
L_{P2}	$= K_w = \frac{\nu}{z_m u_m} = \frac{1}{Re_{mf}}$	1
L_3	$= c_d \cong f_f$	2
L_{M4}	$= A_4 = 1/m_0$	3
L_{P6}	$= \alpha^*_T = \alpha^*_T(u^*, m^*, S^*_w) = \frac{\alpha_T}{u_m c_{pf}\rho_f}$	4
$L_{P7N}(z)$	$= \frac{Pe_m}{A_k(L_{M4}-1)}\alpha^*_T = const$	5

variable	definition	No.
L_{P8}	$= \sigma_b = \frac{K_m}{K_b}$	6
L_{B8}	$= \frac{A_k a_f}{z_m u_m} = \frac{A_k}{Pe_m} = A_k L_{p5} = const$	7
L_{B9}	$= \frac{1}{Pe_{TB}} = K^*_{TB} = \frac{a_{TB}}{z_m u_m}$	8

where $Re_{mf} = \frac{4u_m m_0}{\nu S_{wm}} = \frac{u_m d_{por}}{\nu}$, $Pe_m = PrRe_{mf} = \frac{\nu}{a_f}\frac{d_{por}u_m}{\nu}$, $A_k = \frac{k_s}{k_f}$, $(L_{M4}-1) = \left(\frac{1}{m_0}-1\right) = \frac{s_0}{m_0}$, the ratio of volumetric fractions of solid and fluid phases, $\alpha^*_T(z) = \frac{\alpha_T(z)}{\alpha_{Lm}} = \frac{\alpha_T(z)}{u_m c_{pf}\rho_f}$, $\alpha_{Lm} = \frac{k_{fm}}{z_m} = u_m c_{pf}\rho_f$. Also a_{TB} is the turbulent heat diffusivity coefficient at the boundary porous layer-solid phase, $\widetilde{K}_m$, and K_b are the averaged turbulent eddy viscosity and the turbulent kinetic energy exchange coefficient.

The scale parameters used in the turbulent filtration in this porous medium model are $m_m = m_0$; $z_m = x_m = \frac{4m_0}{S_{wm}}$; $Re_{mf} = \frac{4u_m m_0}{\nu S_{wm}} = \frac{u_m z_m}{\nu}$; $u_m = \left(-\frac{z_m}{\rho_f}\frac{d\langle \overline{p}\rangle_f}{dx}\right)^{1/2}$; $b_m = u_m^2$; $T_m = \frac{z_m Q_0}{K_{Tm}}$; $K_{mm} = z_m u_m = \nu Re_{mf}$; $K_{Tm} = K_{mm}c_{pf}\rho_f = z_m u_m c_{pf}\rho_f$; $S_{wm} = \frac{6(1-m_0)}{d_p}$ or S_{w0}; $\alpha_m = \frac{K_{Tm}}{z_m} = u_m c_{pf}\rho_f$; $c_{dm} = \frac{2u_m^2}{u_0^2}$. Here m_0 is the scale for variable porosity function. It can be given value of a mean porosity in the medium. All of the above transformations used in the nondimensional, turbulent regime governing equations (Travkin and Catton, 1995; Gratton et al., 1996)

$$\frac{\partial}{\partial z^*}\left(K_m^*\frac{\partial u^*}{\partial z^*}\right)=2L_3u^{*2}-L_{M4}, \qquad (21)$$

$$K_m^*\left(\frac{\partial u^*}{\partial z^*}\right)^2+\frac{d}{dz^*}\left(\left(\frac{K_m^*}{L_{P8}}+K_w\right)\frac{db^*}{dz^*}\right)+ \qquad (22)$$
$$+4\left(L_3\right)u^{*3}-2K_w\left(\frac{db^{*1/2}}{dz^*}\right)^2=C_1\frac{b^{*2}}{K_m^*},$$

$$u^*\left(z^*\right)\frac{\partial T_f^*\left(x^*,z^*\right)}{\partial x*}=\frac{\partial}{\partial z^*}\left(K_T^*\left(z\right)\frac{\partial T_f{}^*}{\partial z^*}\right)+$$

$$+4L_{P6}\left[T_s^*\left(x^*,z^*\right)-T_f^*\left(x^*,z^*\right)\right], \qquad (23)$$
$$L_{P6}\ =\ \alpha_T^*\left(z\right)S_w^*\left(z\right)=const, \qquad (24)$$

$$\frac{\partial^2T_s{}^*}{\partial z^{*2}}=4\left(L_{P7N}\right)\left[T_s^*-T_f^*\right], \qquad (25)$$

with the boundary conditions

$$K_m^*\ \mid\ _{z=+0}=K_w=\frac{\nu}{z_mu_m}, \qquad (26)$$
$$-\mathrm{L}_{B8}\frac{\partial T_s^*}{\partial z^*}\ \mid\ _{z=+0}=1,\ -\mathrm{L}_{B9}\frac{\partial T_f^*}{\partial z^*}\mid_{z=+0}=1. \qquad (27)$$

In the laminar regime the governing equations bring the six nondimensional parameters to control the performance of heat sink

Name	variable	minimum	maximum	No.
L_{3N}	$=Re_{mf}c_d$	10^{-5}	5×10^7	1
L_{M4N}	$=Re_{mf}\left(1/m_0\right)$	10^{-3}	10^5	2
L_{P5}	$=\frac{1}{Pe_m}$	2.1	2×10^7	3
L_{P6}	$=\alpha_L^*=\frac{\alpha_L(z)}{\alpha_{Lm}}$	1.0	10^8	4
L_{P7N}	$=\frac{Pe_m}{A_k(L_{M4}-1)}\alpha_L^*$	0	10^{20}	5
L_{B8}	$=\frac{A_k}{Pe_m}=A_kL_{p5}$	10^{-3}	10^{12}	6

These sets of VAT equations for turbulent and laminar regimes we accepted as simulating tool for multiparameter statistical design of experiments for optimization of heat sink design based on two-scale presentation of heat transfer processes (Travkin et al., 2000, 2001c).

One can observe from the above governing statements that even the simple VAT formulations of the scaled device as semiconductor heat sink are in great difference to the homogeneous models. The following question is natural - For what reason we need to complicate the modeling effort ?

And the same simple answer apply - there is no other way to improve the modeling nature and to get improvements in performance - more than 50 years of heat exchangers design evidences in a favor of this movement.

The next logical step in obtaining the better performance characteristics for volumetric heat dissipation device (as heat sink, for example) is to consider the whole number of effects which participate in the momentum and heat transport in a bulk heterogeneous volume. Thus, in the momentum equation those are convective and diffusive heterogeneous fluctuations transport terms in governing equations (Travkin et al. 2000). In fluid temperature equation, for example, those are the terms with convective fluctuations transport and two terms with surficial effect of inhomogeneous temperature of interface. All of these four terms need to be added to conventional heat flux via interface exchange term. For the problems of such a complexity with integro-differential partial differential equations at present time unknown an applicable optimization theory. We applied to this multiparameter optimization problem the method of design of experiments (DOE) (Travkin et al., 2000,2001a). Our results in simulation, experimentation and optimization studies allowed us to design new series of three scale semiconductor heat sink which we will discuss in an outgoing work.

Conclusions

While considering the problem of experimental setups and experimental data reduction for the two-scale semiconductor heat sink the number of new criteria for momentum and heat transport in that problem were derived to connect the local and overall (as temperature in inlet and outlet, etc.) characteristics to a mathematics of VAT scaled models. The heat transfer device is presented as the two scale local- non-local heterogeneous heat exchanger with controls on both scales. For example, the heat transfer rates and effectiveness' are formulated for both phases which improve the energy balance assessment, while characterize better the model using morphology characteristics.

The reason for heterogeneous parameters usage is shown while deploying the analysis of experimental results for heat sink performance - it is the better, more exact description of influence of medium and both phases characteristics on transport performance.

These criteria (parameters) are so specific that allows to distinguish the input of any mechanism or mode of heat transfer occurring in the device. We outline the few levels of modeling available in this problem with consequences for experimental procedures and design, because the bigger number of influencing phenomena make possible the bigger number of choices in optimization of performance or just in increasing the heat exchange rate to its possible highest level. The latter is the goal of preference in cooling of semiconductor devices.

Our experimental results were simulated using nonlocal VAT approach and also compared to a number of works in the area of heat sink design and simulation. All of this helped to direct purposeful design of new class of heat sinks.

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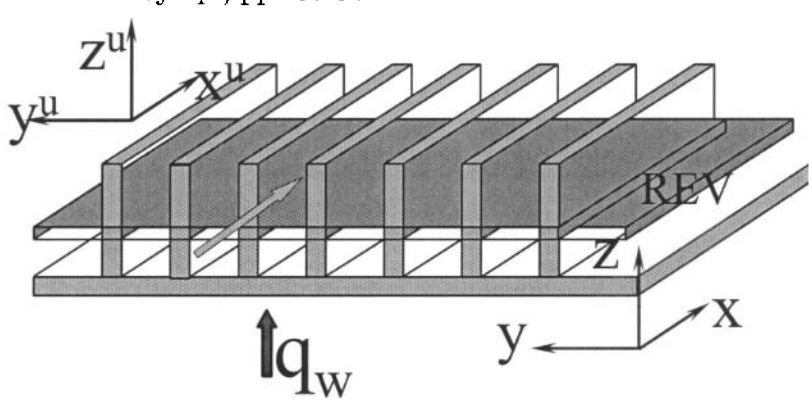
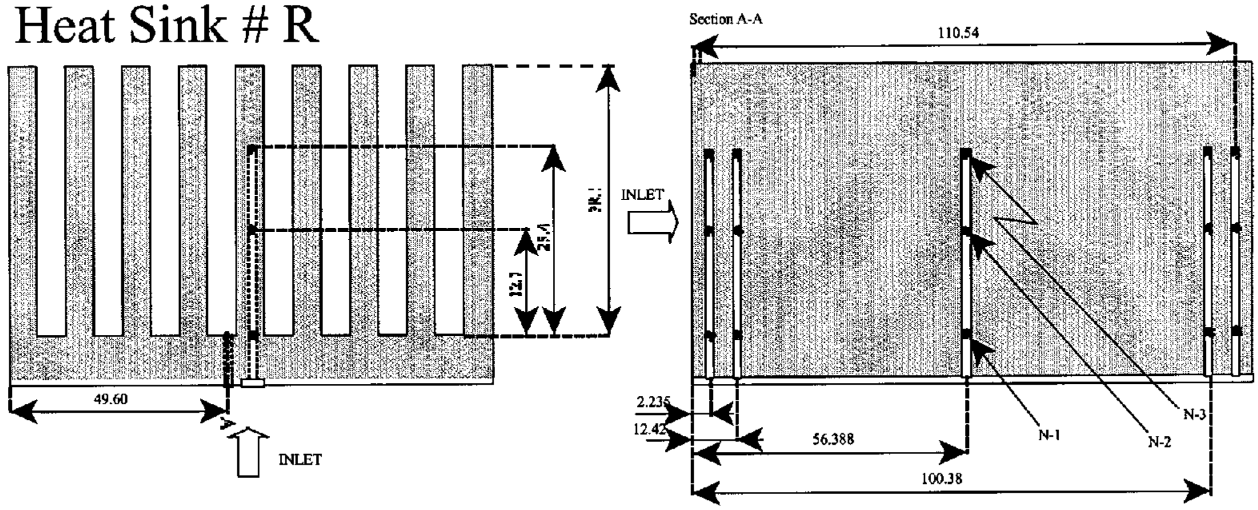


Fig. 1 Longitudinal fins heat sink – relation between lower and upper scale representations, unit vectors are the same but scales are different. This REV is moving along of 0-z^u axis during the averaging and closure calculations

Heat Sink # R



Heat Sink # 3 Characteristics

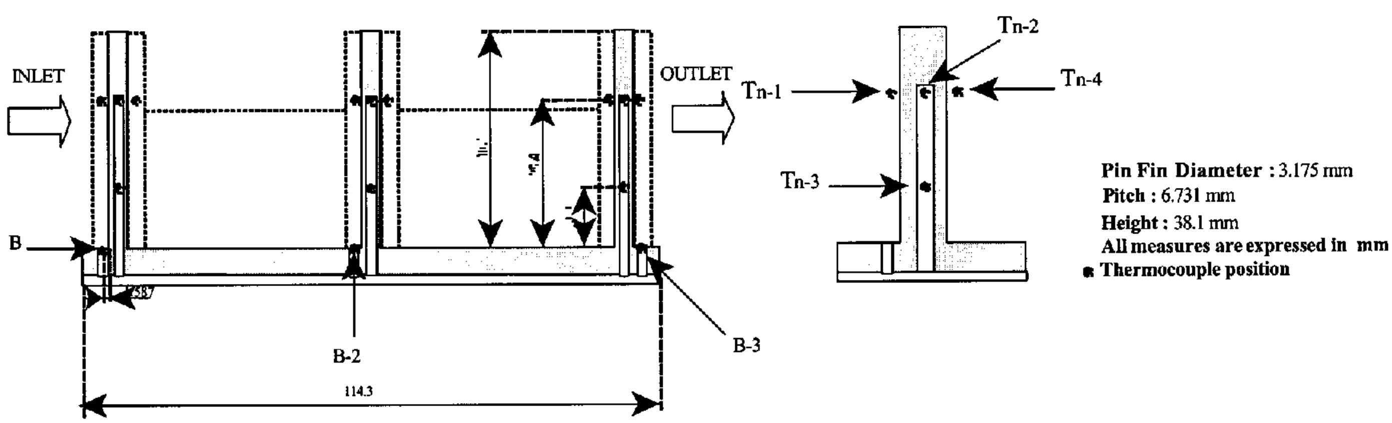


Figure 2 Measurements inside solid phase allowed to assess the heat fluxes

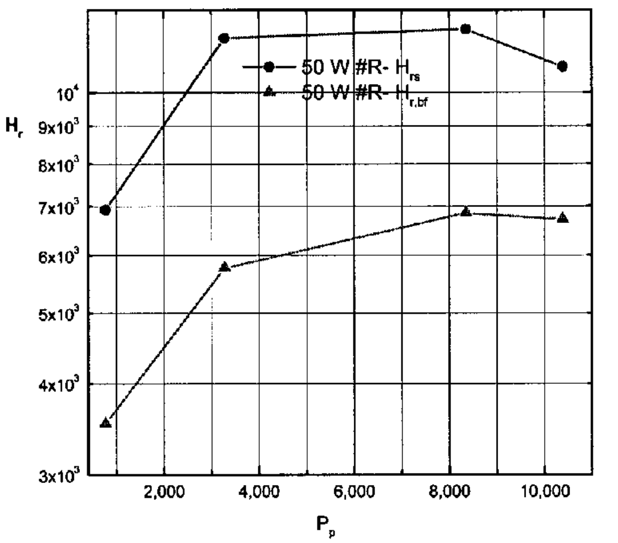


Figure 3 Solid phase heat transfer rate $H_{rs}(P_p)$ and bottom surface (minus fins occupied area) fluid phase heat transfer rate $H_{r,bf}(P_p)$ in experiments with heat sink #R

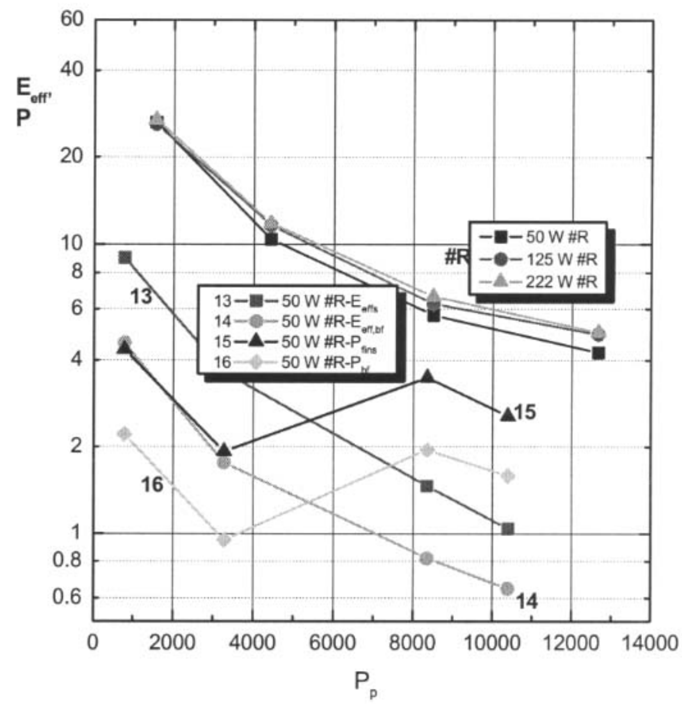


Figure 4 Fluid phase bulk heterogeneous effectiveness $E_{eff1}(P_p)$ for plate heat sink as well as solid phase effectiveness $E_{effs}(P_p)$ and bottom surface (minus fins occupied area) fluid effectiveness $E_{eff,bf}(P_p)$ and dissipated through fins $P_{fins}(P_p)$ and bottom surface $P_{bf}(P_p)$ amount of heat in experiments with heat sink #R