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Integrated Heterogeneous Design of Semiconductor Heat Sink via Scaled Direct Micro-Modeling, Upper Scale VAT Simulation and Experiment. Comparison and Verification of Properties

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Abstract

The thermal and hydrodynamic characteristic of straight-channel longitudinal fin heat sink under forced air-cooling is studied experimentally and numerically using a Phoenics 3.3 (CHAM, UK) CFD software. The aim of the study is to simulate and compare the viability of employing Phoenics code as effective and efficient means gathering the vast amount of data required to adequately quantify heat sink characteristics on the lower scale with the two scale experimental data reduced according to requirements of VAT. For this case is analyzed the basic homogeneous heat transfer performance features and shown insolvency of using heat transfer and effectiveness parameter. A number of data reduction parameters and procedures were developed using scaling heterogeneous formulation by volume averaging theory (VAT). Analysis brought in this work supports the procedures used to connect the two scale VAT experiment with the full lower scale direct numerical modeling for the semiconductor heat sink.

Keywords: heat sink, experiment, data reduction, heterogeneous media, performance characteristics, Phoenics code, grids, model of turbulence

Nomenclature

d_{por}	$=\langle m\rangle/S_w$ characteristic length [m]
E_{effl}	Effectiveness of heat transfer per unit volume [1/K]
f_f	Fanning factor of momentum resistance on the volume [-]
k_f	conductivity coefficient of fluid phase [W/m K]
k_s	conductivity coefficient of solid phase [W/m K]
H_r	heat transfer rate per unit volume per unit temperature difference [W/m^3 K]
L_x	length of heat sink in [m]
$\langle m\rangle$	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 heat sink [W/m^2]
S_{all}	overall heat surface-internal surface plus bottom wall wetted surfase [m^2]
S_w	specific surface of internal volume (bottom surface not included) [1/m]
S^*_w	overall specific surface per unit volume of heat exchanger [1/m]
S_{wint}	internal wetted surface minus bottom surface [m^2]
S_{wb}	bottom wetted surface [m^2]
T_{in}	inlet temperature of coolant [K]
T_{max}	maximum temperature of the wall [K]
$\tilde{\bar{U}}$	averaged interstitial velocity [m/s]
α^*_w	combined (averaged over the all internal plus bottom surfaces) heat transfer coefficient [W/m^{2}K]
$\bar{\alpha}_{int}$	mean heat transfer coefficient in volume of heat sink (averaged over the internal surface) heat transfer coefficient [W/m^2 K]
Ω	volume of heat sink [m^3]
ρ_f	Destiny of coolant fluid [kg/m^3]

Introduction

The one way to achieve the increasing heat transfer rate within a particular volume of heat collector is through the introduction of additional heat exchanging elements (ribs, fins or pins of different shape). At this case the problem becomes a two- scale heterogeneous volumetric heat exchanger design problem. The processes on the lower scale heat transport and around a single transfer element no longer describe the heat transfer rate of the whole sink. At the same time the formulation of the problem of a heat sink for one-temperature or even 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 theoretical scaled (VAT) description and simulation of processes to the experimental variables and to the methodology of experimental set-ups, we came to the processes of coupling of the two- scale Detailed Micro Modeling-Direct

Numerical Modeling (DMM-DNM) using PHOENICS 3.3 code and corresponding experiment for the heat sink design.

Forced-air cooling of surfaces with longitudinal fin heat sink has been attractive technique due to the to its inherent simplicity and cost effectiveness. When the air velocity between the fins of such sink can be well approximated the engineer can effectively predict the overall heat transfer coefficient and pressure drop by applying simple parallel-plate correlation.

The mathematical model

The present work focuses on fluid flow and coupled heat conduction-convection for turbulent flow in rectangular duct with transverse ribs, as shown in Fig.1.

The following set of equations was solved for carrying out the calculation for every cell:

1) continuity equation:

$$\frac{\partial}{\partial x}(\rho u)+\frac{\partial}{\partial x}(\rho v)+\frac{\partial}{\partial x}(\rho w)=0$$

2) movement equation for x, y and z axis respectively:

$$\frac{\partial}{\partial x}(\rho uu)+\frac{\partial}{\partial y}(\rho vu)+\frac{\partial}{\partial z}(\rho wu)=$$
$$-\frac{\partial P}{\partial x}+\frac{\partial}{\partial x}\left[(\mu_l+\mu_t)\frac{\partial u}{\partial x}\right]+\frac{\partial}{\partial y}\left[(\mu_l+\mu_t)\frac{\partial u}{\partial y}\right]$$

$$\frac{\partial}{\partial x}(\rho uv)+\frac{\partial}{\partial y}(\rho vv)+\frac{\partial}{\partial z}(\rho wv)=$$
$$-\frac{\partial P}{\partial y}+\frac{\partial}{\partial x}\left[(\mu_l+\mu_t)\frac{\partial v}{\partial x}\right]+\frac{\partial}{\partial y}\left[(\mu_l+\mu_t)\frac{\partial v}{\partial y}\right]$$

$$\frac{\partial}{\partial x}(\rho uw)+\frac{\partial}{\partial y}(\rho vw)+\frac{\partial}{\partial z}(\rho ww)=$$
$$-\frac{\partial P}{\partial z}+\frac{\partial}{\partial x}\left[(\mu_l+\mu_t)\frac{\partial z}{\partial x}\right]+\frac{\partial}{\partial y}\left[(\mu_l+\mu_t)\frac{\partial z}{\partial y}\right]$$

3) energy equation:

$$\frac{\partial}{\partial x}(\rho uh)+\frac{\partial}{\partial y}(\rho vh)+\frac{\partial}{\partial z}(\rho wh)=$$
$$\frac{\partial}{\partial x}\left[\left(\frac{\mu_l}{Pr_l}+\frac{\mu_t}{Pr_t}\right)\frac{\partial h}{\partial x}\right]+\frac{\partial}{\partial y}\left[\left(\frac{\mu_l}{Pr_l}+\frac{\mu_t}{Pr_t}\right)\frac{\partial h}{\partial y}\right]$$

4) the ordinary k-e model:

$$\nu_t=C_\mu\frac{k^2}{\varepsilon}$$

Turbulent kinetic energy:

$$\frac{\partial}{\partial x}(\rho uk)+\frac{\partial}{\partial y}(\rho vk)+\frac{\partial}{\partial z}(\rho wk)=$$
$$\frac{\partial}{\partial x}\left[\left(\mu_l+\frac{\mu_t}{\sigma_k}\right)\frac{\partial k}{\partial x}\right]+\frac{\partial}{\partial y}\left[\left(\mu_l+\frac{\mu_t}{\sigma_k}\right)\frac{\partial k}{\partial y}\right]+\mu_t\Phi-\rho\varepsilon$$

Dissipation rate:

$$\frac{\partial}{\partial x}(\rho u\varepsilon)+\frac{\partial}{\partial y}(\rho v\varepsilon)+\frac{\partial}{\partial z}(\rho w\varepsilon)=$$
$$\frac{\partial}{\partial x}\left[\left(\mu_l+\frac{\mu_t}{\sigma_\varepsilon}\right)\frac{\partial\varepsilon}{\partial x}\right]+\frac{\partial}{\partial y}\left[\left(\mu_l+\frac{\mu_t}{\sigma_\varepsilon}\right)\frac{\partial\varepsilon}{\partial y}\right]$$
$$+C_1\mu_t\frac{\varepsilon}{k}\Phi-\rho C_2\frac{\varepsilon^2}{k}$$

where Φ - dissipation coefficient, $\sigma_k=1.0$, $\sigma_\varepsilon=1.3$, C_1=1.44, C_2=1.92, $C_\mu=0.09$.

Boundary conditions

At the inlet (west) $u=u_{in}$, $h=h_{in}$, $k=k_{in}$, $\varepsilon=\varepsilon_{in}$,

where $k_{in}=0.0025(u_{in})^2$ and $\varepsilon_{in}=\frac{1.6(k_{in})^{1.5}}{Re_{por}}$

At the outlet (east) gradient of all variables set as zero.

The low wall is exposed to a uniform heat flux q_w and the other three boundaries are adiabatic.

Wall function

The wall function concept is based on experimental observation that velocity profiles for turbulent flow fall on single curve when plotted in terms of inner variables. At some distance from the wall, turbulent effects dominate the molecular contribution and u^+ can be represented by a logarithmic function of y^+,

$$u^+=\frac{1}{\chi}\ln(Ey^+)$$

where u and y denote velocity components on normal distance from the wall respectively and E is constant, χ – constant (0.4).

Applying the above velocity profile at y=1 (near-wall grid point) gives the following equation for SKIN:

$$SKIN^{-\frac{1}{2}}=\frac{1}{k}\ln\left(E\cdot Re_1\cdot SKIN^{\frac{1}{2}}\right)\text{ where}$$

$$SKIN=\frac{\tau_w}{\rho\cdot u_1{}^2}$$

Solving the previous equation for SKIN, the wall shear stress can be computed in such a way that:

$$\tau_w=\rho\cdot u_1{}^2\cdot SKIN$$

To observe the local profile for temperature:

$$T^+=Pr_t\cdot(u^++P)$$

where P is the sublayer resistance function. For smooth walls:

$$P=9\cdot\left(\frac{Pr_l}{Pr_t}-1\right)\cdot\left(\frac{Pr_l}{Pr_t}\right)^{-\frac{1}{4}}\text{ for }Pr_t\geq 0.5,$$

where subscripts "l"- laminar, "t" – turbulent.

When the wall heat flux is given the wall temperature:

$$T_W=\frac{q_W}{\rho Cp\cdot u_1\cdot STAN_1}+T_1$$

where subscript "1"-for the first cell, STAN – is internal Phoenics function.

Numerical model

The ducted heat sink assembly is shown in Fig.1.

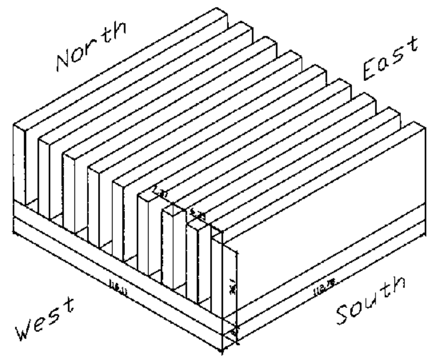


Fig. 1 Ducted Heat Sink Geometry

Air flow direction from West to East (along axis X). Base dimensions is shown in Fig.2,3.

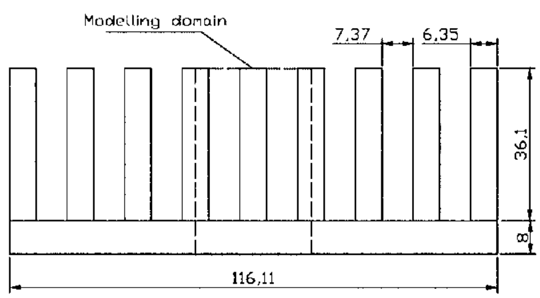


Fig. 2 Sink Dimensions

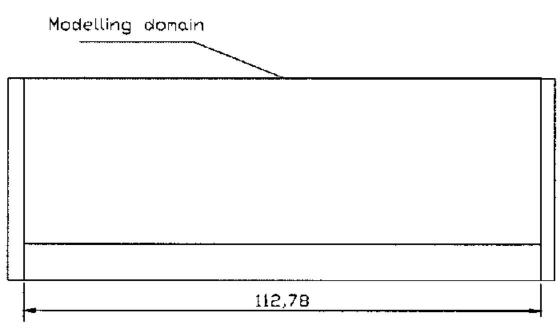


Fig. 3 Sink Dimensions

Some of testing results for flows in flat channel with using PHOENICS were in works [7,8]. Were getting testing results for different geometry of channels flat and with adjustable roofs. For flat channels the results for pressure drops compare with data [9]. The sink was at 1 m from the inlet of channel. Pressure drop according [9] equl 43 Pa and Phoenics 40 Pa. All physic and geometry parameters for channel with adustible roofs corresponds for data [1].

The heat sink configuration described in test matrix was modeled in Phoenics 3.3 code from CHAM (UK). The approach makes use of local distribution of velocities, shear stress and other turbulence quantities in the near-wall region to calculate effectiveness and pumping power.

A computational domain is shown in Figs.4-5.

Rectangular profile was used at the inlet of channel and at the distance 0.9m (0.1 m before sink) was closed to developed profile in flat channel. The grid mesh is x:100, y:100, z:30. To get a stable and converged solution, we need in 10,000 iterations. This required 32h on CPU Pentium III 1Ghz.

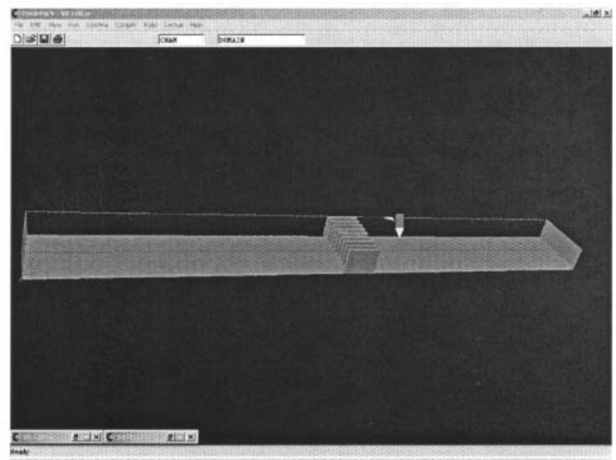


Fig. 4 Channel Geometry

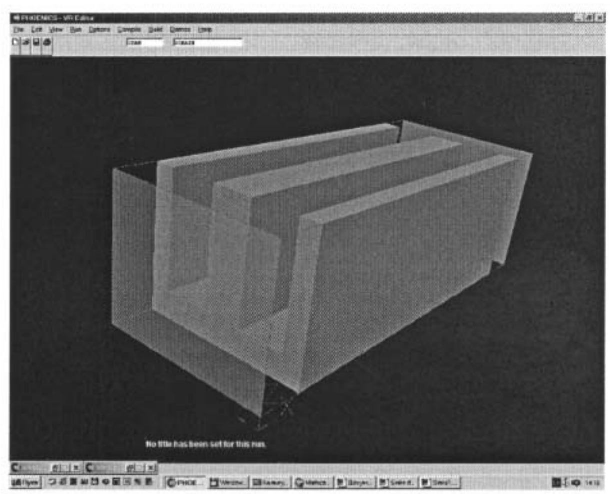


Fig. 5 Computational domain

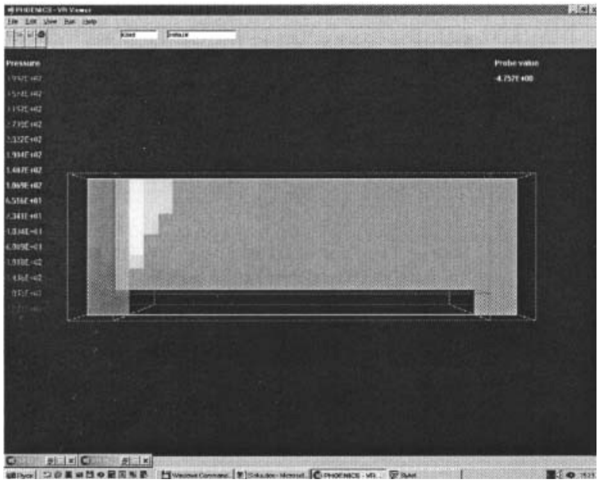


Fig. 6 Pressure Distribution

Heterogeneous VAT Based One Phase Analysis of Heat Sink Modeling Data

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 following:

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

$$H_r = S_{all}\alpha_w^* / \Omega \quad \left[\frac{W}{m^3 K}\right],$$

where S_{all} is the total internal surface, α_w^* is the combined (averaged over the all internal and bottom surfaces) heat transfer coefficient, and Ω is the volume of heat transfer; and

b) pumping power per unit volume of sink P_p

$$P_p = \frac{P}{\Omega}, \quad \left[\frac{W}{m^3}\right],$$

c) effectiveness parameter for the volume of heat sink that results in

$$E_{eff1} = \frac{H_r}{P_p}$$

These characteristics formulae given above are the same as used for heat exchangers performance elsewhere. Using the two scale VAT approach their detailed mathematical formulations would be very different as disclosed below.

To calculate the above parameters one needs to find first the bulk (mean) Fanning friction factor f_f for the volume of the heat sink which can be assessed using formulae based on VAT for experimental measurements of pressure loss (see Travkin an Catton, 1998; Travkin et al., 1999):

$$f_f = \left[\frac{2\langle m\rangle \Delta p}{\rho_f \tilde{U}^2 S_w L_x}\right],$$

where $\tilde{U}$ is average interstitial bulk velocity estimated for the volume where heat transfer occurs. The pumping power per unit volume for a heterogeneous media is given by following:

$$P_p = \frac{P}{\Omega} = \frac{\dot{m}\Delta p}{\rho_f \Omega} = f_f Re_{por}^3 \langle m_{yx}\rangle \left(\frac{S_w^4}{\langle m\rangle^4}\right)\frac{\mu^3}{128\rho_f^2}, \quad \left[\frac{W}{m^3}\right]$$

where

$$\langle m_{yz}\rangle = \frac{S_{yz}}{L_y L_z}, \quad L_z = H,$$

A second, but more important characteristic to evaluate the heat exchange device, is the heat transfer rate $H_r = \frac{S_{all}\alpha_w^*}{\Omega}$ for known heat flux q_w through the bottom surface of heterogeneous volumetric devices used as heat exchangers

$$H_r = \frac{S_{all}\alpha_w^*}{\Omega} = Nu_w \frac{k_f S_w}{4\langle m\rangle} S_w^*, \quad \left[\frac{W}{m^3 K}\right]$$

from which the Nusselt number can be found. The Nu_w here is not an internal porous medium heat transfer Nusselt number averaged across both phases. Implicitly it is – the overall (bottom and internal surface) Nusselt number

$$Nu_w = \frac{q_w d_{por}}{(T_{w\,max} - T_{in})k_f}$$

The ultimate parameter for most kinds of heat exchangers is ratio of energy transfer rate to pumping power, H_r/P_p, which is the effectiveness of heat transfer per unit volume per unit temperature difference. For a heterogeneous volumetric two-scale heat transfer device it is:

$$E_{eff1} = \frac{H_r}{P_p} = \left[\frac{Nu_w}{f_f Re_{por}^3}\left(32\frac{S_w^*}{\langle m_{yz}\rangle}\frac{\langle m\rangle^3}{S_w^3}\right)\frac{k_f \rho_f^2}{\mu^3}\right], \quad \left[\frac{1}{K}\right]$$

Based on DMM-DNM data were analyzed based on given above characteristics and compared with previously calculated parameters obtained via physical modeling of the same longitudinal fin heat sink (Rizzi et al. 2001; Travkin et al., 2001) We've got very closely located data, which are plotted in in Figs. 7-10. Most of the analysis regarding experimental results can be found in mentioned above works.

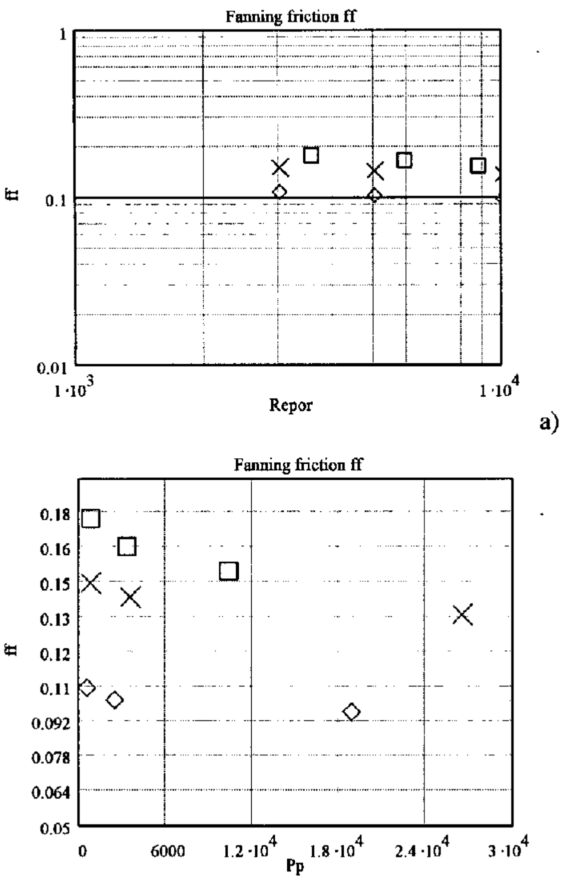


Fig. 7. Fanning friction factor ff: X-DMM-DNM (based on CFD modeling), Box-Experiment, Rhombus-with by-pass 5mm

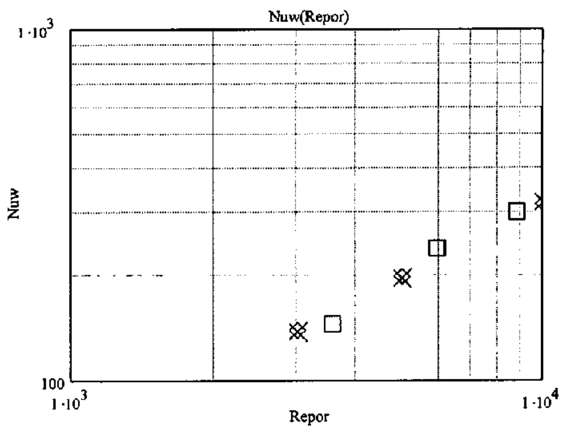


Fig. 8 Scaled Nusselt number Nu_w through the bottom cross-section

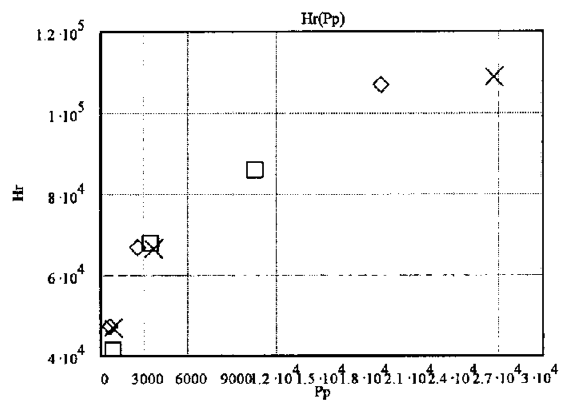
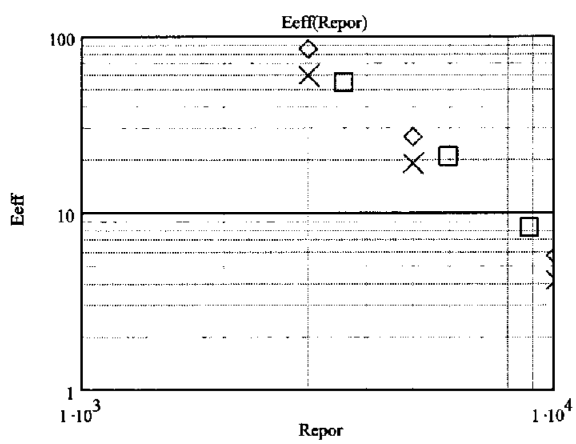
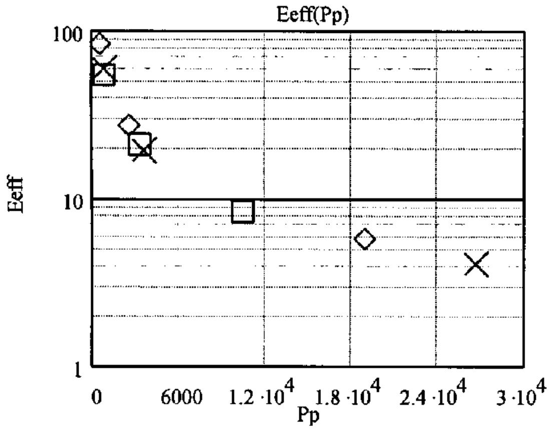


Fig. 9. Comparison of heat heat transfer rate for simulation.



a)



b)

Fig. 10 Effectiveness E_{effi} of heat transfer in the plate fin sink

Conclusion

The scaled volume average theory (VAT) description and simulation of semiconductor heat sinks heat transfer (heat sink-to-air) to experimental measurements were developed along with coupling of two scale Detailed Micro Modeling – Direct Numerical Modeling (DMM-DNM) and their corresponding experimental result for a heat sink design.

The proper set-ups for the measurement methods and used the local and bulk variables data to the corresponding simulation procedures in the VAT upper scale governing equations were used in measurement over two classes of heat sinks.

Four sample semiconductor heat sinks of two morphologies were experimentally studied by different techniques and models. The result were depicted 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 VAT based formulation for the heat transfer rate, fin effectiveness, and influence of only morphology features among others.

The nondimentional parameters of friction factor f_f, Nu_w, and the effectiveness E_{effi}, computed with PHOENICS appeared to be in a good accordance with the experiment (Rizzi et al, 2001; Travkin et al., 2001). Lower scale direct numerical modeling of the longitudinal fin heat sink provided with the use of CFD code to study the data reduction procedures and result of the physical experiment. The result of simulation shown that the PHOENICS code is the good tool for analysis of local and averaged. characteristics in VAT turbulent two scale simulations as, for example for the heat sink. The design of some new candidate heat sink surfaces has been completed.

Acknowledgments

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