HEAT TRANSFER THROUGH THE COMPONENTS OF CPU'S HEAT SINK

Pătuleanu Liliana[1], Manolache-Rusu Ioan-Cozmin[2], Andronic Florin[3], Radion Ivan[4]

[1,2,3] *A Department of Mechanical Engineering, Stefan cel Mare University of Suceava, 13 University Street, zip code:720229 Suceava, Romania, [1] lilianapatuleanu@yahoo.com, [2] cozmin.manolache@yahoo.com, [3] foiand@yahoo.com, rad_ivan2005@yahoo.com*

Abstract: *The necessity of higher processing speeds has determined the evolution of technology in the field of expertise of computer science. There have been manufactured CPUs that need increasingly higher power, thus there have been applied new methods and built complex systems in order to solve the cooling problem. The thermal management of the computer imposes close supervision of CPU's temperature in different working conditions.*

The current article aims to realize a brief presentation of air cooling systems and the concerns regarding the improvement of heat transfer through the heat sink. In the second part, it is realized a calculation model of heat transfer through the components of the heat sink, in order to optimize its geometry. The used model is validated by comparison with numerical results obtained by authors in specialized literature.

Keywords: *heat transfer, heat sink, mini and microchannels.*

1. Introduction

The thermal management is designed to measure CPU's temperature in different conditions of use, to evaluate the reliability and to identify the possible defects caused by overheating. Due to the fact that intense CPU usage may endanger its integrity and result in a lower lifetime of the entire system, there have been conceived cooling systems that allow the control of the maximum value a CPU can rise up to.

For the evaluation of the temperatures reached by a CPU in different conditions of use (normal, intense and overload), there was performed an experiment on an ASUS N53SN Notebook equipped with Intel Core i7-2630QM of 2.0 GHz in an indoor environment recording a stable temperature of 18,5^{o}C. The computer had a Windows 7 Ultimate installed (64 bits version) and for a more accurate measurement there were no unnecessary applications running in system background, which could determine an unwanted rise in temperature.

The actual measurement of the core temperature was recorded using the software Open Hardware Monitor [1], which gather data from the internal CPU thermal sensor, thus for creating an overload situation there was developed a software, which opens threads in infinite loops by using a recursive function without an end condition. The obtained results for the four CPU cores in a time interval of 5 minutes of recordings were represented in the diagrams from Fig. 1.

Minimum solicitation corresponds to the situation in which the computer is just opened and there is no user intervention afterwards. Also, there are no applications open other than the ones meant to keep the system running and responsive.

The intense solicitation of the CPU is obtained by opening the program, data indicating a significant rise in temperature, from an average value of 40°C to 57°C.

Overload imposes using the CPU at its maximum capacity, a situation where all of the four cores are 100% loaded. The maximum value reached is 70°C.

In Fig. 1 can be observed an abrupt change in temperature of all the four cores. The significant change with $\Delta T = 30°C$ is produced when the active programs perform a high number of tasks in a short interval of time. As a precautionary measure for avoiding such high temperature that can result in an irreversible deterioration of the CPU, it is imposed to use an efficient cooling system and to continuously monitor the temperature.

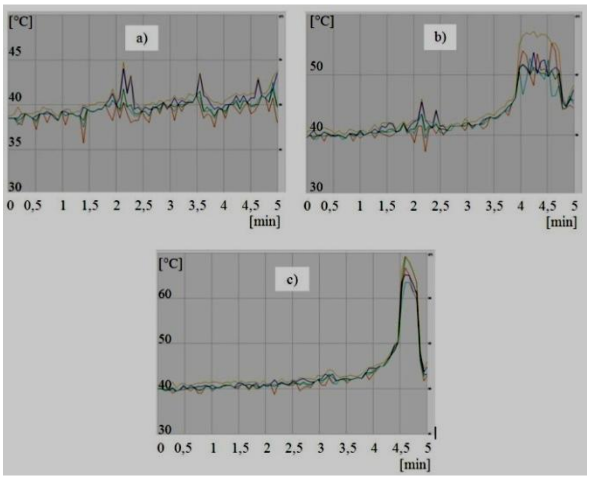
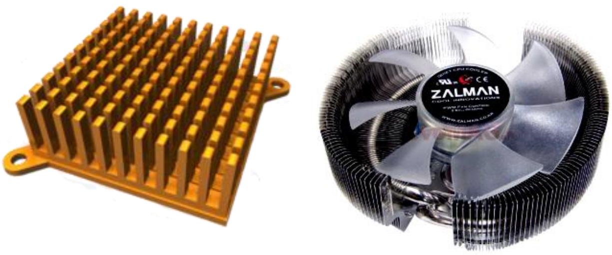


Figure 1: *The evolution of temperature, in different conditions of solicitation, in 5 minutes. a–minimal, b–intense, c –overload.*

2. Considerations regarding air cooling of the CPU

The evolution of CPU's performances and also the miniaturization of the electronic components have been made possible only by adopting technical solutions designed to ensure the dissipation of the generated heat. There have been discovered and continuously improved more and more techniques to evacuate the thermal flux generated by the electronic components. The first cooling systems were the ones using air, passive and also active. The active heat sink is the main cooling method of CPUs and graphic boards.



a. Passive heat sink, [17]. *b. Active heat sink, [18].*
Figure 2: *Types of heat sinks.*

The efficiency and reliability of air cooling systems depend on the thermal performance of the package CPU – cooling heat sink. In order to increase the maximum dissipated power value, it is necessary that the total thermal resistance is reduced. The assembly of a heat sink allows a more efficient heat transfer from the source – the CPU, because of the existence of "extended surfaces". The study of these CPU cooling components appeals to the manufacturers and users of these systems. In Fig. 2 are presented types of heat sinks and fins.

The efficiency of the heat sink is given by the material of which it is composed, the cooling surface, the shape and the contact surface with the CPU.

Air cooling is next applied on a large scale for the microelectronic components with reduced thermal charge densities. Ever since the apparition of this cooling technology [2], there has been studied the geometry of the heat sink (the width of the channel, the thickness and shape of the fins), and [3] optimized its thermal performances. Up until the present time, there have been proposed different calculation models for average speed, thermal resistance, pressure losses and temperature distributions. The calculations are necessary in determining the fluid flow rate and heat transfer in the microchannels of the heat sink, in a given configuration, in case of natural convection [4, 9, 10] and forced convection
[3, 5, 6, 7, 8, 11, 12].
The solutions were validated by comparison with the results gathered in numerical simulation and experiments.

P. Holman studies, in [4], natural convection of a heat sink made of extruded aluminum and proposes a calculation model for the rise in temperature in the casing of the heat sink, in a known configuration.

Kim, in [3], conceived a method of optimization using an averaging approach for the study of forced convection in cooling fluid flowing in the direction of the microchannels of the heat sink, with the upper surface layer isolated and the lower surface layer uniformly heated. In the case of mathematical formulation of the problem, there are made the following simplifying hypothesis: the fluid flow rate is laminar, incompressible, hydrodynamic and completely thermal developed; the power dissipated is considered constant; the aspect ratio of the channel is considered higher than 1; the conductivity of the solid material is higher the conductivity of the liquid.

In paper [6] is analyzed the circulation of the cooling fluid in the case of force convection, signaling that "a high part of the air emitted by the heat sink can take the minimum resistance path and flow around the heat sink, producing a bypass flow". It is proposed the calculation model of air flow rate through the heat sink and the thermal resistance and there are drawn graphs that show the accentuated rise of thermal resistance at bypass flow, for a high number of fins. The same author computes, in [5], the pressure drop in a heat sink with volumetric air flow rate. The results of the calculations show that: when the number of fins rises, the air velocity rises, but the thermal resistance drops.

The calculation of the heat transfer coefficient, pressure drop, heat sink thermal resistance, depending on the number of fins with different heights and thicknesses, horizontal or vertical, for different geometries of the heat sink [3, 5, 10, 11], highlights the configuration in which the thermal resistance is minimal.

The calculation of the joint temperature according to the number of fins, from [7], indicates the optimum heat sink design conditions.

For a certain number of fins, it will be produced a low temperature of heating and above this number, an excessive heating dangerous for the CPU might appear.

The studies prove that in a CPU cooling application with heat sink, are important the following: the way in which the heat transfer is done (conduction, natural or forced convection, radiation), the geometric characteristics of the heat sinks used (their fin number, their location), the position of the heat source, and the thermal conductivity, quality and reliability of the thermal grease used as a heat exchange environment.

3. The calculation of heat transfer inside the heat sink

The thermal transfer from the CPU – the thermal source concentrated on the surface, through the heat sink and to the environment, is carried out by conduction, convection, radiation or a combination of two or all three of the phenomena.

We propose to analyze the heat transfer for a CPU dissipating a power of 95 W, through a heat sink, in the case of forced air cooling.

By applying the calculation model proposed in [5, 8] and developed in [15], considering a heat sink with horizontal minichannels, Fig. 3, oriented in the direction of air flux, with the following characteristic dimensions: the width of the heat sink w = 0,060 m; the length of the heat sink L = 0,060 m; the height of the heat sink h_{hs} = 0,037 m; the number of fins N_f = 20; the height of fins h_f = 0,030; the height of the mainboard h_b = 0,007; the thickness of the fins t_f = 0,0015 m.

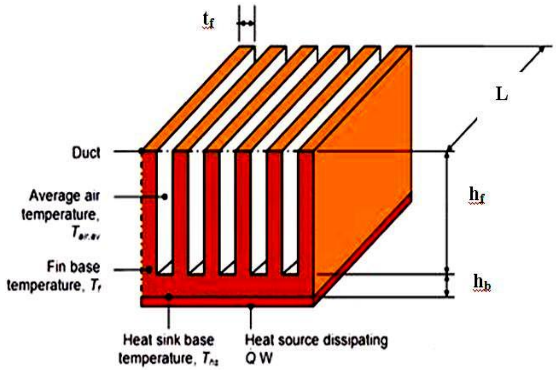


Figure 3: *The geometry of the heat sink,[16].*

The chosen model is resolved in MathCAD for the following values: the kinematic viscosity of air nu = 1.5 · 10^{-5}; the air density ρ = 1,2 kg/m^3 ; the thermal capacity of air at constant pressure (kJ / kg · K) C_p = 1; the loss coefficient K = 0,9; the room temperature measured in the computer T_a = 316 K; the maximum CPU temperature T_b = 358 K; the Prandtl for air Pr = 0,7; the volumetric flow rate of air at Cooler Master is G = 2.36 · 10^{-3} m^3/s; the thermal conductivity of Al, k_{al} = 170 W/m·K; the thermal conductivity of air k_{air} = 0,026 W/m·K.

For configuring the heat sink with the presented parameters, it is calculated:

- b displacement between fins:

$$b = \frac{w - N_f \cdot t_f}{N_f - 1} \tag{1}$$

- A_b base surface between fins:

$$A_b = \left(N_f - 1\right) \cdot b \cdot L \tag{2}$$

- A_f surface area of the fins:

$$A_f = 2h_f \cdot L \tag{3}$$

- A surface on which the conductive heat exchange is done;
- P perimeter of the surface;
- D_h hydraulic diameter: $D_h = \dfrac{4 \cdot A}{P}$ (4)

The thermal resistance of the heat sink is calculated using the relation, from [11]:

$$R_{hs} = \frac{1}{h \cdot \left(A_b + N_f \eta_f A_f \right)} \qquad (5)$$

In which: N_f is the number of fins; η_f is the efficiency of the fin.

The efficiency of the fin is established using relation (6) from [14]:

$$\eta_f = \frac{\left[e^{(m \cdot h_f)} - e^{-(m \cdot h_f)} \right]}{\left[e^{(m \cdot h_f)} + e^{-(m \cdot h_f)} \right] m \cdot h_f} \qquad (6)$$

with:

$$m = \sqrt{\frac{2 \cdot h}{k_f \cdot t_f}} \qquad (7)$$

The variation of η_f according to N_f is represented in Fig. 4 as follows:

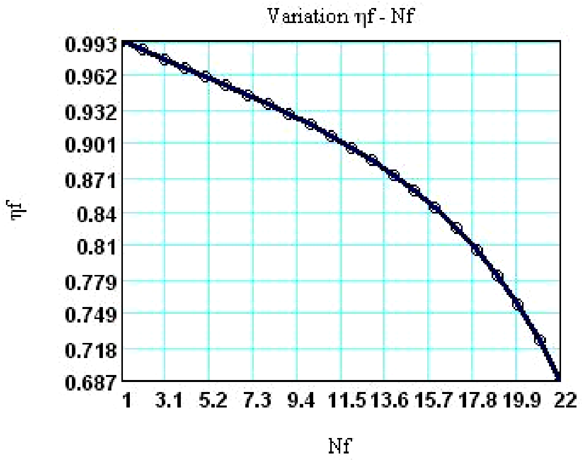


Figure 4: *Variation η_f – Nf.*

The convective heat transfer coefficient h, is calculated using relation (8):

$$h = Nu_b \frac{k_{fluid}}{b} \qquad (8)$$

where: k_{fluid} is the thermal conductivity of air; Nu_b is the Nusselt number calculated with relation (9), from [12]:

$$Nu_b = \left[\frac{1}{\left[\frac{\mathrm{Re} \cdot \mathrm{Pr}}{2} \right]^3} + \frac{1}{\left[0{,}664 \sqrt{\mathrm{Re}} \cdot \mathrm{Pr}^{0{,}33} \cdot \sqrt{1 + \frac{3{,}65}{\sqrt{\mathrm{Re}}}} \right]^3} \right]^{-0{,}33} \qquad (9)$$

The graphical representation of the convective heat transfer coefficient h, according to the number of fins N_f = 1-22 is presented in Fig.5.

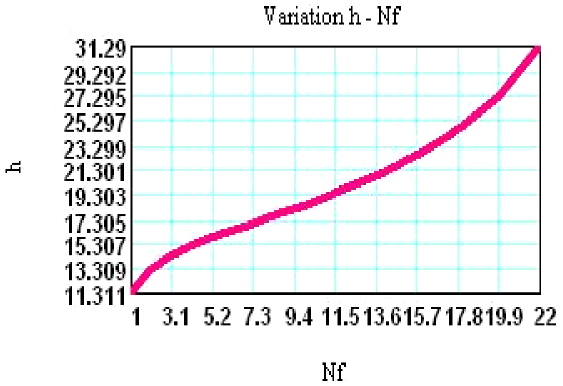


Figure 5: *Variation h – Nf.*

The Nusselt number in the flow direction is calculated using relation (10) and is presented in Fig. 6:

$$Nux = 0{,}453 \cdot \mathrm{Re}\, x^{\frac{1}{2}} \cdot \mathrm{Pr}^{\frac{1}{3}} \qquad (10)$$

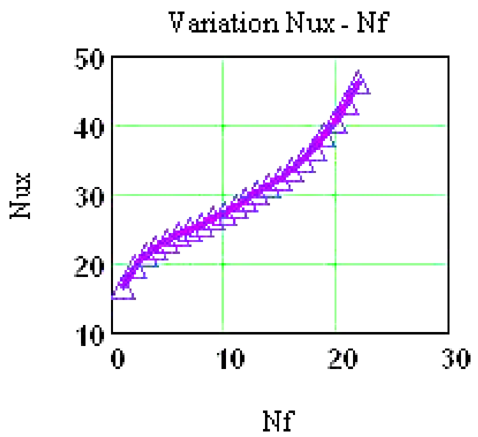


Figure 6: *Variation Nux – Nf.*

The calculation of the total thermal resistance, R_{tot}, presumes to add to the convection resistance calculated before, the thermal conductive resistance at the base of the heat sink (k_b the thermal conductivity of the base). For the uniform heat flux, the total thermal resistance $R_{tot,}$ [11] represented in Fig. 7, is given by:

$$R_{tot} = R_{hs} + \frac{h_{hs} - h_f}{k_b \cdot w \cdot L} \qquad (11)$$

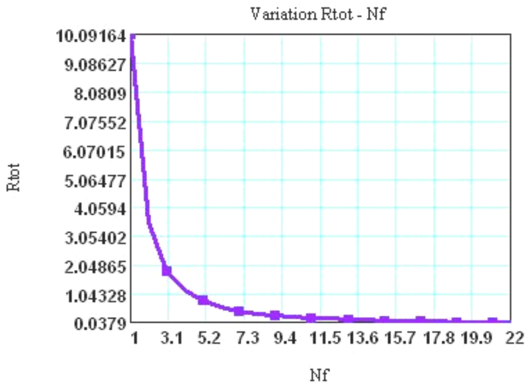


Figure 7: *Variation R_{tot} – Nf.*

The heat flux transmitted through the heat sink is evaluated using the relation:

$$Q_w = \frac{P}{\left(N_f \cdot 2 \cdot h_f + w\right) \cdot h_{hs}} \tag{12}$$

The graphical representation of Q_w is given in Fig. 8 and shows that the heat flux is reduced according to the rise in the number of fins.

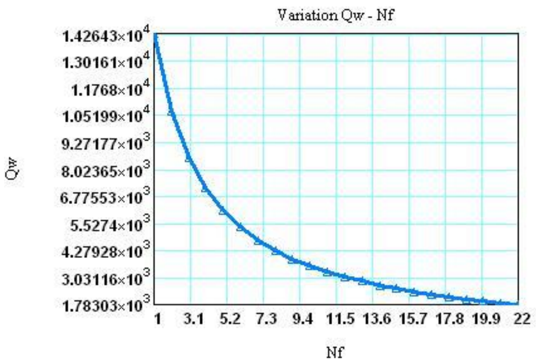


Figure 8: *Variation Q_w – Nf.*

In order to compare the variation of the convective heat transfer coefficient h, with the variation of the heat flux Q_w depending on the fin number N_f, there are drawn the graphs in Fig. 9.

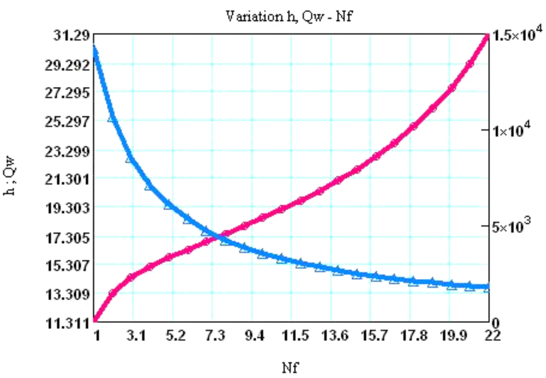


Figure 9: *Variation h, Q_w – Nf.*

4. Conclusions

The calculation models presented allow the optimum choice of heat sink's geometry that offers the maximum efficiency of the cooling system.

The graphical representation of the variation of the convective heat transfer coefficient depending on the number of fins
$N_f = 1\ldots 22$ is presented in Fig. 5 and show the rise of the convective heat transfer coefficient, together with the rise of the number of fins, which means that the process of heat exchange is intensifying by utilization of a higher number of fins.

The variation of the efficiency of the fins η_f is represented in Fig. 3, shows that the efficiency of the fins drops according to the rise in the number of fins.

The obtained results and the representations from Fig. 6, 4, 7, 8, 9 show that the thermal resistance, the heat transfer coefficient and the thermal flux through the heat sink have close values and the same way of variation as the one obtained by other authors in [5, 7, 11, 13, 15].

In this way, the obtained results are validated.

ACKNOWLEDGMENT

This paper was supported by the project "Improvement of the doctoral studies quality in engineering science for development of the knowledge based society-QDOC" contract no. POSDRU/107/1.5/S/78534, project co-funded by the European Social Fund through the Sectorial Operational Program Human Resources 2007-2013.

References

[1] http://openhardwaremonitor.org/O. H. Monitor.org, "*Open Hardware Monitor*".

[2]D.B. Tuckerman, R.F.W. Pease,*High-performance heat sinking for VLSI*, IEEE Electr. Dev. Lett. 2. - 1981.

[3]Kim, D-K. and Kim, S. J., "*Closed-form correlations for thermal optimization of microchannels*,", International Journal of Heat and Mass Transfer, Vol. 50, No. 25-26, Elsevier, 5318–5322 - 2007.

[4] Holman J.P., *Calculation of Temperature Rise of Shrouded Heat Sink Under Natural Convection*,Heat Transfer, pg. 42. . - 2008.

[5] Simons R.E. , *Estimating Parallel Plate-Fin Heat Sink Pressure Drop*, Electronics Cooling - 2003.

[6] Simons R.E. - *Estimating The Effect Of Flow Bypass On Parallel Plate-Fin Heat Sink Performance*,Electronics Cooling - 2004.
[7]Azar K. ş.a. - How Much Heat can be Extracted from a Heat Sink? ElectronicsCooling - 2003.
[8] Guenin B., *Packaging Challenges For High Heat Flux Devices,* Electronics Cooling, 2006.
[9] Simons R.E. – *Simplified Formula for Estimating Natural Convection Heat Transfer Coefficient on a Flat Plate*,Electronics Cooling - 2001.
[10]Simons R.E. – *Estimation of natural convection heat transfer for vertical parallel plates flat panels,* Electronics Cooling - 2002.
[11] Simons R.E. , *Estimating Parallel Plate-Fin Heat Sink Thermal Resistance (159-163)*, Electronics Cooling - 2003.
[12]Teerstra, P., Yovanovich, M.M., and Culham, J.R., *Analytical Forced Convection Modeling of Plate Fin Heat Sinks*, Proceeding of 15th IEEE Semi – Therm Symposium, pp. 34-41, 1999.
[13] Wei, J., *Challenges in Cooling Design of CPU Packages for High-Performance Servers*, Heat Transfer Engineering, 29(2): 178-187, 2008
[14] Chiriac F., A. Leca, M. Pop, A. Badea, L. Luca, N. Antonescu, D. Peretz,*Processes of heat and mass transfer in the industrial facilities*, Ed. Tehnică, Bucharest, 1982.
[15] Mihai, I., *Heat Transfer in Minichannels and Microchannels CPU Cooling Systems* | InTechOpen,Published on: 2011-01-28.www.intechopen.com, p. 64-68.
[16] http://en.wikipedia.org/wiki/Heat_sink
Heat sink,
[17] http://en.wikipedia.org/wiki/Intel_GMA, *Intel GMA*,en.wikipedia.org/wiki/Intel_GMA, 2012.
[18] http://www.evomag.ro/COMPONENTE-PC-Coolere-Coolere-CPU/Zalman-Cooler-procesor-CNPS-8700-NT.html