Sensors and Materials, Vol. 14, No. 8 (2002) 429–441
MYU Tokyo

S & M 0502

Computer Simulation of a Pin-Fin Heat Sink with Fluid Cooling for Power Semiconductor Modules

Igor Khorunzhii[1*], Hubertus Gabor[1], Reinhart Job[1], Wolfgang R. Fahrner[1] and Heinrich Baumann[2]

[1]Chair of Electronic Devices, University of Hagen, D-58084 Hagen, Germany
[2]Bombardier Transportation Company, D-68309 Mannheim, Germany

(Received September 24, 2002; accepted December 16, 2002)

Key words: heat, temperature, cooling, thermal resistance, heat sink, volume flow rate

The steady state forced-convective cooling of a heat sink for a power (~1 kW heat power) semiconductor module with staggered pin-fin combination has been investigated on the basis of computer simulations. A commercial software package ABAQUS(1) is used for the development of the computational model. A self-developed subprogram for the calculation of the local heat transfer coefficient between solid and liquid is used. The effect of the thermal conductivity variation of the sink material, volume flow rate of the cooling liquid, total heat power and inlet cooling liquid temperature has been investigated. Dependencies of the thermal resistances, calculated on the bases of highest and average temperatures of the heating surface as a function of the thermal conductivity of the pin-fin sink material, heat power, liquid volume flow rates, and liquid inlet temperature have been obtained. The impacts of the thermal conductivity of the pin-fin sink material and volume flow rate of the cooling liquid on the thermal resistance of the cooling system are shown.

1. Introduction

For many industrial applications, internal heat generation can cause serious overheating problems and sometimes lead to system failure. According to a U.S. Air Force study,(2) the four primary sources of stresses that cause failures in avionics systems are temperature (~55%), vibration (~20%), excessive humidity (~19%), and dust (~6%). Thus, the temperature is the most critical parameter for ensuring the reliability of electronic systems. For a microelectronic device, every additional 10°C increase above 65°C approximately halves its mean-time-to-failure.(3) To overcome this problem, thermal systems with com-

*Corresponding author, e-mail address: igor.khorunzhii@fernuni-hagen.de

pact efficient heat sinks (e.g., pin-finned heat exchangers) and forced convection are used.[4,5] At the same time, the design of a pin-fin heat sink is a complicated technical problem due to the complicated geometry, the need for detailed calculation of the heat transfer coefficient and the complicated manufacturing process. Computer simulation is a suitable tool for this investigation because it allows studying the influence of every parameter on the cooling system properties. At the same time, we have found only a few articles with descriptions of computer simulations of a pin-fin heat sinks with natural[6] and forced convection.[7,8] The models of pin-fin heat sinks with forced convection presented in these articles are based on a self-developed computer code and are rather complicated to reproduce. The model presented in this paper based on the standard commercial software package ABAQUS and can be repeated rather easily by skilled ABAQUS users.

2. Computer Model Description

A computer-generated design of the heat converter is presented in Fig. 1. This heat converter is used as a cooling system for powerful (about 1 kW heat power) semiconductor modules. There are two main parts to the converter: a pin-fin heat sink and a channeling box. The material of the pin-fin sink is an AlSiC ceramic with high thermal conductivity (about 180 $Wm^{-1}K^{-1}$). The sink has dimensions of $137 \times 127 \times 5$ mm^3. There are 928 cone-like pins from one side of the sink, which are staggered over the rectangular area of 95.6×108 mm^2. A schematic of the pin distribution at the pin-fin sink and specific geometrical

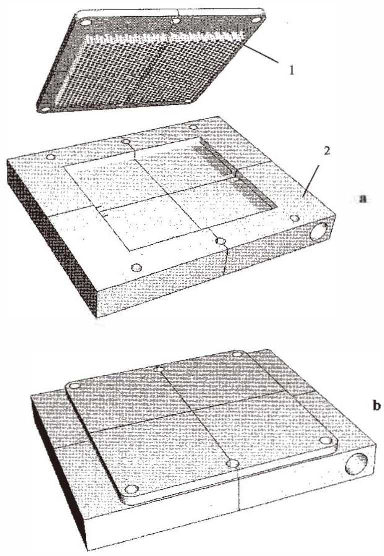

Fig. 1. Computer generated design of the cooling system for power semiconductor module: a) Disassembled heat converter: 1 - pin-fin sink, 2 - channeling box; b) Assembled heat converter.

parameters are presented in Fig. 2. The length of the pins is 6 mm, their diameters are 2.4 mm at the foot and 1.8 mm at the top, and *a* and *b* parameters are 3.586 mm and 3.106 mm, respectively. The pin-fin sink is joined to a channeling box. Exactly in front of the pin region, the channeling box has a recess with dimensions of 96 × 108.5 × 6.5 mm^3; this recess is connected to an inlet and an outlet by a system of channels. The material of the channeling box is a special plastic with low thermal conductivity (about 0.29 $Wm^{-1}K^{-1}$). In operating mode, the volume inside the converter is filled with moving cooling fluid. Thus, the main heat transfer process in the converter is a forced convection. In our model, we consider each part as consisting of four quarters. The reasons for such a consideration are the following: 1) the complicated geometry of the converter, especially of the pin-fin plate, creates a situation where ABAQUS has insufficient prescribed memory to retain information about the whole single part; 2) with such a consideration, we are able to study temperature distributions and heat flux distributions across the cross section of the converter. ABAQUS has no possibility to generate a cross section of a single part or get information about heat fluxes or temperature distribution across the cross section of a single part. During computer simulation all parts of our computer model are considered together with ideal heat contacts between quarters. It is worth noting that the finite element mesh must be fine enough, which means that lateral dimensions of finite elements must be at least a few (3–5) times smaller than the specific dimensions of the converter geometry. Such a fine mesh is necessary for a correct simulation of the heat exchange in the pins. Computer-generated meshes for one quarter of the pin-fin sink and one quarter of the channeling box are presented in Fig. 3. Finite element meshes of similar quality are generated for the other quarters of the model. As a result, the computer model includes about 415000 finite elements and needs a minimum of 675 MB system memory (the optimal memory size is 1.42 GB). In order to check the mesh quality we have done one calculation with significantly finer mesh. In this calculation the number of finite elements was increased by about 50% compared with our usual calculation and had reached the number of about 600000. The finest mesh was generated for pins. The maximal temperature in this calculation was only lower by 0.08 degrees than the maximal temperature calculated with our usual mesh for the same initial parameters. Thus, we concluded that the mesh quality is good enough.

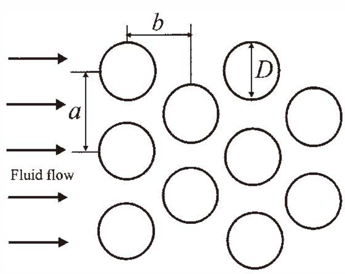

Fig. 2. Schematic of pin distribution at the pin-fin sink and specific geometrical parameters.

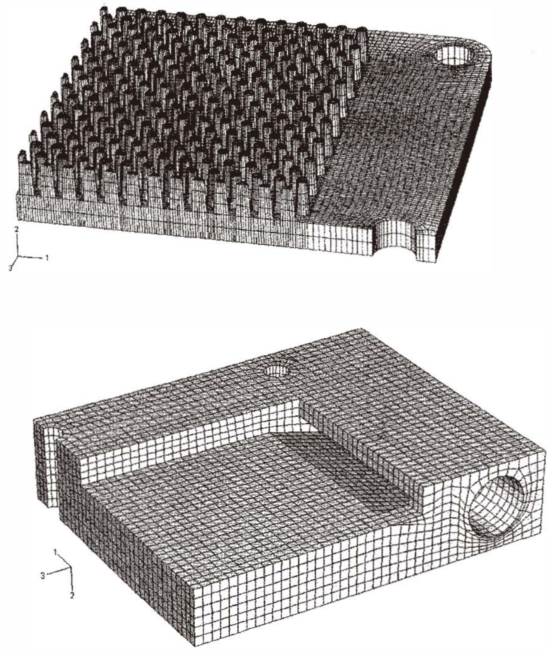

Fig. 3. 3D-finite element mesh (FEM) generated for the pin-fin heat sink (a) and for the channeling box (b).

As has been mentioned above, the main heat exchange takes place due to forced convection of the cooling fluid. For fluid cooling simulation, a so-called "film" condition (a boundary layer model) is used on the contact solid-liquid interface. Newton's law of cooling describes the heat exchange between a solid and liquid[9,10]:

$$q = h \cdot (T_w - T_f), \tag{1}$$

where q is the heat flux per unit of area, h is the heat transfer coefficient, and T_w and T_f are the temperatures of the solid wall and the cooling fluid, respectively. The bulk fluid temperature T_f is obtained by considering the temperature of the coolant at the inlet to the duct T_0 and the rate of heat addition Q up to the point of interest[9]:

$$T_f = T_0 + \frac{Q}{V \cdot \rho \cdot c_p}, \tag{2}$$

where V is the volume flow rate, ρ is the liquid density and c_p is the specific heat of the coolant at constant pressure.

The most complicated problem is the calculation of the heat transfer coefficient *h*. This coefficient depends on many parameters such as volume, flow rate, density and viscosity of liquid, characteristic of flow (laminar or turbulent), specific geometrical parameters of the system and so on. ABAQUS has no tool for fluid dynamic calculations (CFD); therefore in our model, an empirical equation from refs. (10,11) is taken for the heat transfer coefficient *h* calculation:

$$Nu_f = C \cdot 0.35 \cdot \left(\frac{a}{b}\right)^{0.2} Re_f^{0.60} Pr_f^{0.36} \left(\frac{Pr_f}{Pr_w}\right)^{0.25}, \tag{3}$$

where *Nu* is the Nusselt number, *a* and *b* are specific geometrical parameters (see Fig. 2), and *Re* and *Pr* are the Reynolds and Prandtl numbers, respectively. Subscripts *f* and *w* denote which temperature (of the fluid (*f*) or the solid wall (*w*)) has to be used for the number calculation. The average diameter of pin *D* is used for *Nu*, *Re* and *Pr* number calculation. As is noted in ref. (10), the factor $(Pr_f/Pr_w)^{0.25}$ allows us to take correctly into account the change of properties of the cooling liquid in a boundary contact layer. The heat transfer has different efficiencies for pin rows with different numbers of pins. For the first fourteen rows, it increases with row number due to increasing turbulence of flow.(11) The factor *C* is used for taking into account the pin row number. The dependence of the C-factor on the row number is presented in Fig. 4. Equation (3) is valid for $100<Re<2 \cdot 10^5$.

The heat transfer coefficient may be calculated easily from eq. (3) by means of the following equation:(9)

$$h = \frac{Nu_f \cdot \lambda_f}{D}, \tag{4}$$

where λ_f is the thermal conductivity of a cooling liquid.

A self-developed subprogram in the algorithmic language Fortran-90 has been written for the calculation of the local value of the heat transfer coefficient. At every element of the solid-liquid contact surface, the heat flux *q* (eq. (1)) from the pin-fin sink to the coolant is

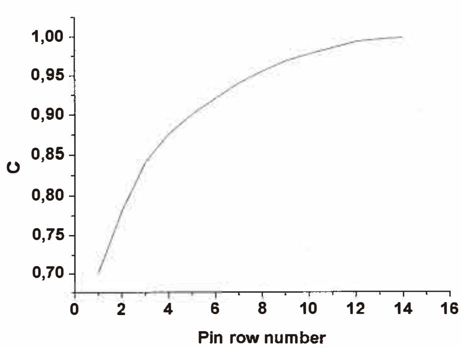

Fig. 4. Dependence of C-factor on the row number [8].

calculated while taking into account the local value of the solid wall temperature, the local bulk cooling fluid temperature and the local heat transfer coefficient. As boundary conditions, we use the given inlet cooling liquid temperature and the given heat flux distribution across a 100×110 mm^2 area across the top (external) surface of the pin-fin heat sink. For our simulation, we have used a uniform distribution of the heat flux across a 100×110 mm^2 area as shown in Fig. 5.

With an Intel 933 MHz processor Pentium III (with 1 GB system memory) one calculation takes about 3 hours.

3. Results of Computer Simulation and Discussion

In Figs. 6 and 7, results of the computer simulation are presented for the heat converter with the following parameters: heat power is 1 kW (uniformly distributed over the area of 100×110 mm^2, see Fig. 5), volume flow rate of cooling liquid is 5 l/min, thermal conductivity of the pin-fin sink is 180 Wm^{-1}K^{-1} (AlSiC ceramic), thermal conductivity of the channeling-box is 0.29 Wm$^{-1}\cdot$K^{-1} (plastic), and inlet temperature of the cooling liquid is 20°C. The parameters of Antifrogen N produced by Clariant[12] (44%-glycol, 56%-water) are used for the cooling liquid. All results are obtained for the steady state regime. The highest calculated temperature at the upper surface of the pin-fin sink is 31.40°C. The average temperature calculated for the area across which the heat flux is distributed is 29.55°C. The outlet temperature of the coolant is 23.25°C. For integral characterization of the cooling system, the thermal resistance TR of the cooling system is calculated by means of the following equation:

$$TR = \frac{T^* - T_0}{P}. \tag{5}$$

Here T^* is the temperature of the upper surface of the pin-fin sink, T_0 is the inlet temperature of the cooling liquid and P is the heat power. We have calculated two thermal resistances

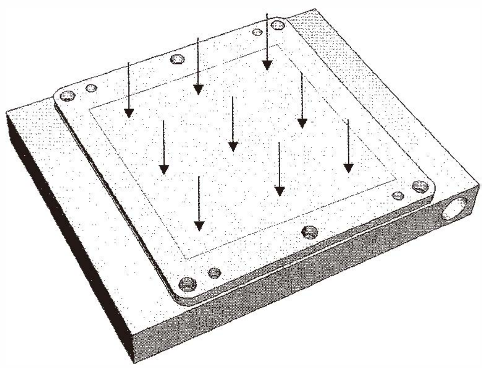

Fig. 5. Boundary condition: heat flux distribution at the top surface of the pin-fin sink.

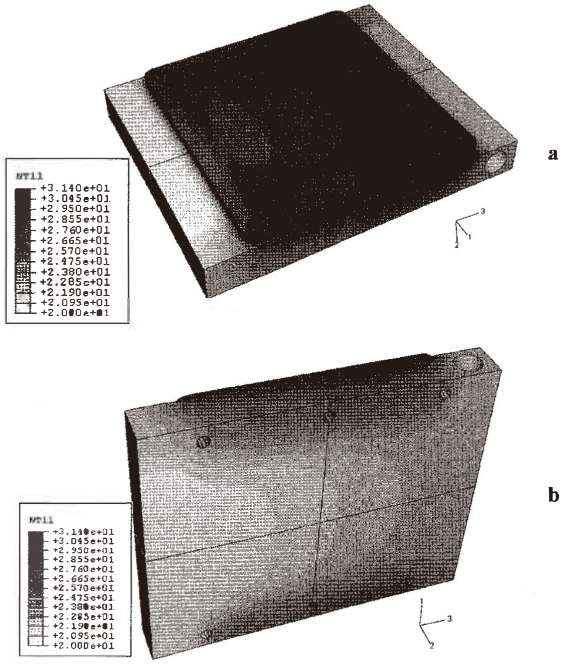

Fig. 6. Temperature distribution across the top surface (a) and the back surface (b) of the heat converter for the stationary regime.

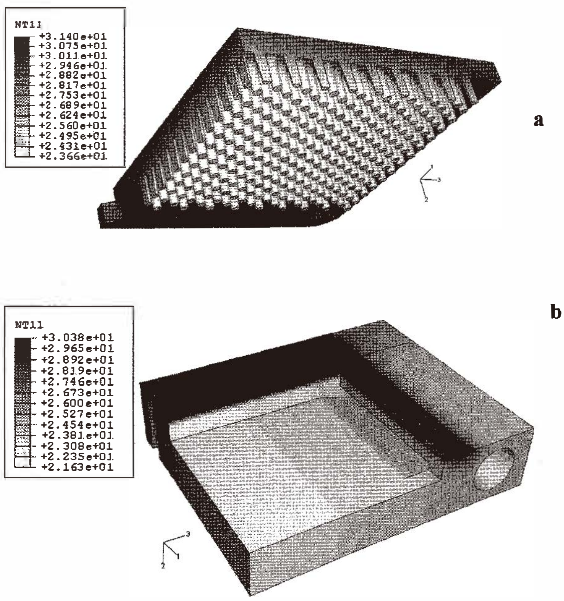

Fig. 7. Temperature distributions across the section and the bottom surface of the pin-fin sink (a) and across the section and the top surface of the channeling box (b).

of the cooling system. The first one is calculated when T^* is the highest temperature of the upper surface, T^*_{max}, and the second one when T^* is the average temperature, T^*_{av}, across the area where the heat flux is distributed. These resistances calculated for the above-mentioned simulation have values of $1.14 \cdot 10^{-2}$ and $9.55 \cdot 10^{-3}$ K/W, respectively.

In Fig. 8, the thermal resistances of the cooling system are presented as functions of the volume flow rate of the cooling liquid. The volume flow rate has a strong impact on the final thermal resistance, because it controls the turbulence of flow and the parameters of the boundary contact layer. The outlet temperature of the coolant depends on the flow rate, too. Results of simulation show that the volume flow rate has the most dramatic influence on the final thermal resistance at lower flow rates (especially at volume flow rates less than 5–6 l/min). There are two main reason for this impact. The temperature of the cooling liquid increases quickly at a volume flow rate less than 5–6 l/min and additionally, the heat transfer coefficient decreases.

In Fig. 9, the thermal resistance of the cooling system is presented as a function of the thermal conductivity of the pin-fin sink material. The thermal conductivity of the pin-fin sink is varied over wide ranges. The thermal conductivity of the pin-fin sink has a strong impact on the thermal resistance of the heat converter, but when the thermal conductivity of the pin-fin sink reaches a value of about 180–200 $Wm^{-1} \cdot K^{-1}$, its further impact on the thermal resistance of the cooling system decreases. Thus, if the thermal conductivity of the pin-fin sink reaches a value of about 180–200 $Wm^{-1} \cdot K^{-1}$, the solid-liquid contact layer makes the main contribution to the thermal resistance of the heat converter. Further reduction of the thermal resistance of the converter is feasible mainly by changing the coolant properties, geometry of the pin design and volume flow rate.

For practical applications, the inlet temperature is usually in the region of 55 to 65°C and, for obvious reasons, a protection against winter freeze-up must be provided. We have studied the impact of the inlet cooling liquid temperature (Fig. 10) and the total heat power (Fig. 11) on the thermal resistance of the converter. Both of these parameters have a

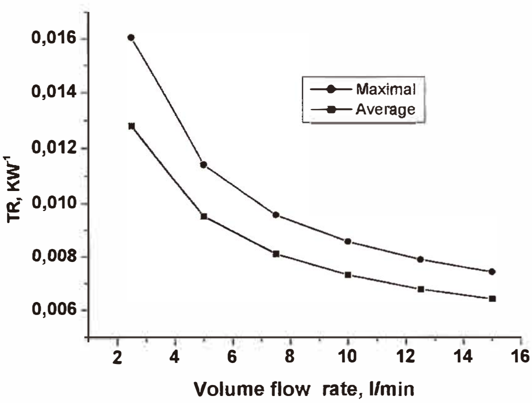

Fig. 8. Integral thermal resistance of the heat converter as a function of cooling fluid volume flow rate. Thermal resistances are calculated by means of eq. (5). Maximal for $T^* = T^*_{max}$, and average for $T^* = T^*_{av}$.

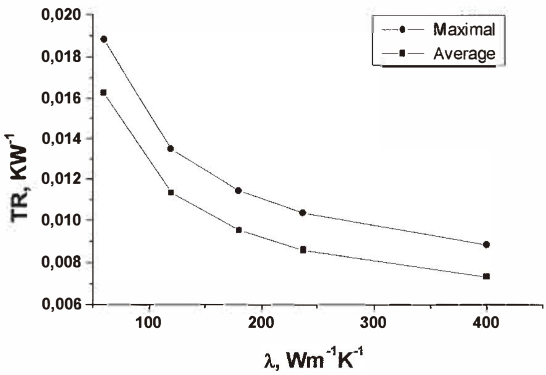

Fig. 9. Converter thermal resistance as a function of pin-fin sink material thermal conductivity.

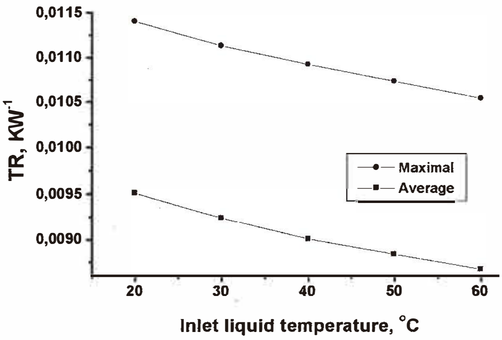

Fig. 10. Impact of inlet cooling fluid temperature on final thermal resistance of the converter.

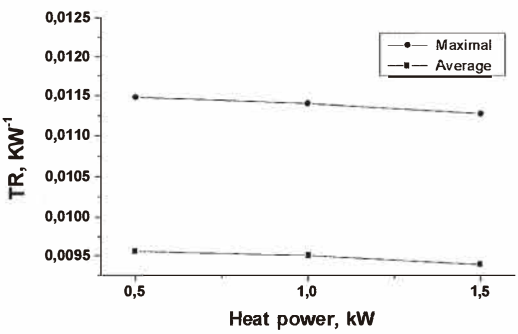

Fig. 11. Impact of heat power on final thermal resistance of the converter.

relatively small impact on the thermal resistance.

This means that the pin-fin heat converter with fluid cooling is some type of auto-regulated system. The increase in temperature immediately leads to an increase in the heat output to the coolant. The dependence is nonlinear due to a strong temperature dependence of the heat transfer coefficient. Due to such auto-regulated behavior, the thermal resistances of the converter for different temperatures of the inlet cooling fluid have values close to each other. Some reduction of the thermal resistance with growth of the inlet temperature takes place due to a reduction of the viscosity of the cooling liquid with temperature growth. This conclusion is confirmed by results of computer simulation and thermal resistance calculation for various values of heat power. The total heat power has a very small impact on the final thermal resistance.

For the reduction of the thermal resistance of the heat converter, different cooling liquids may be used. For studying the impact of the liquid properties on the thermal resistance behavior, we have compared the results of computer simulations with parameters of pure water and with parameters of cooling liquid antifrogen N produced by Clariant.[9] For these calculations, we have used the following parameters: 1 kW heat power and 5 l/min volume flow rate. A comparison of parameters for both liquids at the temperature of 20°C is presented in Table 1.

Due to the higher specific heat, thermal conductivity and lower viscosity, pure water has a better efficiency than the cooling liquid antifrogen N. The thermal resistances calculated for the heat converter with antifrogen N and pure water have the following values: $1.14 \cdot 10^{-2}$ KW^{-1} and $9.59 \cdot 10^{-3}$ KW^{-1} for $T^* = T^*_{max}$ (see eq. (5)), and $9.51 \cdot 10^{-3}$ KW^{-1} and $7.55 \cdot 10^{-3}$ KW^{-1} for $T^* = T^*_{av}$, respectively. Thus, if the antifrogen N (glycol - 44%) is replaced by pure water with a mass flow rate of 5 l/min, the same result may be obtained as if increasing the volume flow rate of the antifrogen N from 5 l/min up to 7–9 l/min. Unfortunately, for practical applications, pure water cannot be used, as the cooling system needs to be made frost-resistant.

It is interesting to compare the results discussed above with results obtained for a simple model. This simple model includes only one pin with a proportional part of the plate (see Fig. 12). As boundary conditions for this model we use the heat flux on the top surface of the plate and film conditions for surfaces having contact with a cooling fluid. At the other surfaces of the model, the condition of thermal isolation is used. The heat flux density used for the simple model has the same value as that of the heat converter simulation. The cooling fluid temperature for the simple model is equal to the temperature of cooling liquid at the center of the converter. For the heat transfer coefficient $\boldsymbol{h}$, calculations from eqs. (3) and (4) are used.

Table 1
Parameters of antifrogen N (glycol-44%), and pure water at 20°C.[9]

Parameter	Antifrogen N (44%)	Pure water
Density, kg·m^{-3}	1072.71	998.55
Specific heat, J·kg^{-1}K^{-1}	3442.47	4186.16
Thermal conductivity, W·m^{-1}·K^{-1}	0.43	0.6
Viscosity, m^{2}s^{-1}	$3.55 \cdot 10^{-6}$	$1.07 \cdot 10^{-6}$

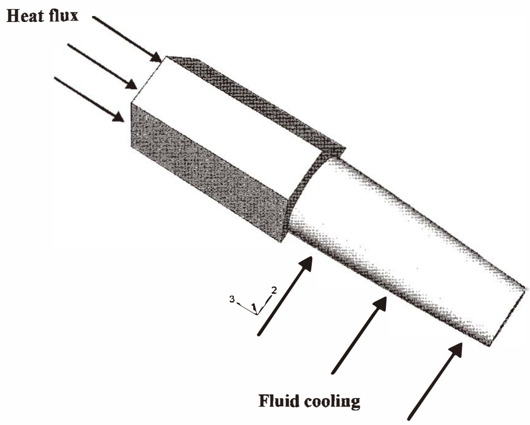

Fig. 12. Design of the simple model.

Figures 13 and 14 show, respectively, the temperature and the heat flux magnitude distributions across the cross section of the simple model for the stationary regime. The temperature on the top surface in the simple model has almost the same value as the temperature at the center of the top surface for the entire model (see Fig. 6). This means that the simple model may be used to study the temperature behavior in the converter. The strongest heat exchange takes place in the upper part of the pin (see Fig. 14) because the maximum temperature difference T_w–T_f is present here. The lower part of the pin makes a smaller contribution to the heat exchange. Figure 15 shows the temperature difference T_{max}–T_0 as a function of the pin length. The behavior of this temperature difference gives us an impression about the thermal resistance behavior. The diameter of the pin in this calculation is fixed. All calculations are carried out at the volume flow rate of 5 l/min. According to this result, the length of pins in the considered pin-fin heat sink has an optimal value. For further improvement of the cooling efficiency, the diameter and number of pins may be changed. But it is necessary to take into account that with the increase of pin diameter, the number of pins (on the fixed area), contact area sink-water and heat transfer coefficient h will be reduced. For a smaller diameter of pins, the heat transfer coefficient h will have a higher value but the effective length of the pin will be reduced.

4. Conclusion

A computer model of the heat converter with fluid cooling for a power (~1kW heat power) semiconductor module is developed on the basis of the commercial software package ABAQUS. An integral thermal resistance of the converter is calculated. The impact of the variations of the thermal conductivity of the pin-fin sink material, volume flow rate of cooling liquid, inlet liquid temperature and total heat power variation have been investigated. If the thermal conductivity of the pin-fin sink material reaches a value of about 180–200 $W \cdot m^{-1} \cdot K^{-1}$, its further impact on the final thermal resistance of the

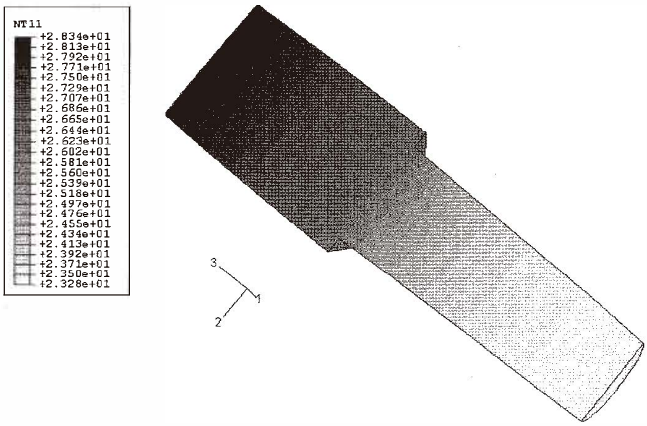

Fig. 13. Temperature distribution across the cross-section of the simple model.

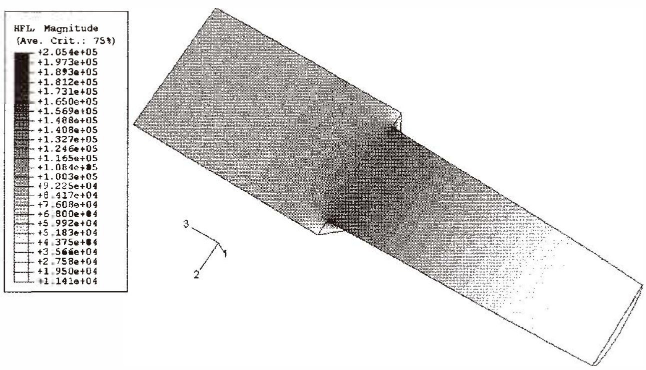

Fig. 14. Heat flux magnitude distribution across the cross-section of the simple model.

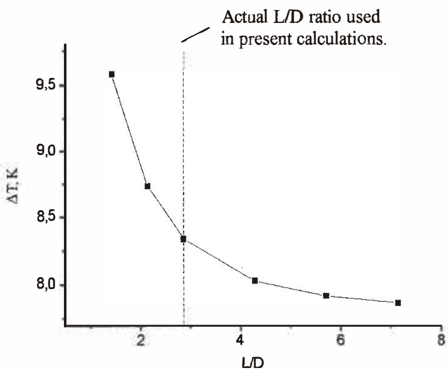

Fig. 15. Temperature difference between maximal temperature on the top surface of the simple model and inlet temperature of the cooling fluid T_{av}–T_0 as a function of the ratio L/D (length of pin divided by its diameter).

converter decreases. The volume flow rate has the most dramatic influence on the final thermal resistance at lower volume flow rates (less than 5 l/min) because the temperature of the cooling fluid increases rapidly. The total heat power and inlet cooling liquid temperature have only a small impact on the final thermal resistance of the converter. For our consideration (with uniform distribution of the heat flux across the top surface of the pin-fin heat sink), maximal temperatures on the top surface obtained by computer simulation of a single pin are in good agreement with average temperatures of the top surface for the whole pin-fin heat sink. Nevertheless, for other types of heat flux distributions across the top surface of heat sink, additional investigations must be done.

Acknowledgments

This work is supported by VDI/VDE-IT, Berlin-Teltow, and the Ministry of Education and Research (BMBF) of Germany.

References

1 ABAQUS is a registered trademark of Hibbit, Karlsson & Sorensen, Inc. USA, 1080 Main Street Pawtucket, RI 02860-4847.
2 M. Reynell: Computer Aided Eng. J. **7** (1990) 104.
3 R. F. Babus'Haq, K. Akintunde and S. D. Probert: Int. J. Heat and Fluid Flow **16** (1995) 50.
4 H. Baumann, P. Heinemeyer, W. Staiger, M. Töpfer, K. Unger and D. Müller: IEEE Transactions on Industrial Electronics **48** (2001) 298.
5 M. A. Tahat, R. F. Babus'Haq and S. D. Probert: Appl. Energy **48** (1994) 335.
6 S. Narasimhah and J. Majdalani: IEEE Trans. Compon. Packaging Technol. **25** (2002) 78.
7 W. W. Lin and D. J. Lee: Int. J. Heat Mass Transfer **40** (1997) 1937.
8 J. G. Maveety and H.H. Jung: Int. Comm. Heat Mass Transfer **27** (2000) 229.
9 R. H. S. Winterton: Heat Transfer (Oxford University Press, Oxford, New York, 1997).
10 A. A. Zukauskas: Convective transfer in heat exchange systems (Science, Moscow, 1982) Chap. 13. (in Russian).
11 S. S. Kutateladse: Heat transfer and hydrodynamics resistance (Atomic Energy Publishing House, Moscow, 1990) Chap. 9. (in Russian).
12 Antifrogen N: Clariant GmbH, Surfactants Division, Functional Fluids/Marketing, D-65840 Sulzbach, Germany.