IMPINGING FLOW PARALLEL PLATES HEAT SINKS

Valter Cesar de Souza *[pastor@fem.unicamp.br]*
Carlos A. C. Altemani *[altemani@fem.unicamp.br]*

State University of Campinas
Faculty of Mechanical Engineering - Department of Energy
CP 6122 – ZIP CODE 13.083-970 – Campinas – São Paulo

Abstract. Microcomputers performance has been increased with higher electric power dissipation within it. Atmospheric air is the most convenient fluid for thermal control but, due to its properties, it usually requires the use of enhanced heat transfer devices. Heat sinks are the most widely used and their thermal design needs a careful work. The most standard heat sink configuration employs parallel plates and a forced fluid flow along the heat sink base. In an alternative configuration, the airflow impinges over the top of the heat sink and it deviates to the two lateral sides of each parallel plates channel as it approaches the heat sink base. The convective heat transfer and the flow head loss are larger in this configuration than in the former one. In the present work, three parallel plates heat sinks were built and tested with variable airflow under the conditions of top inlet and side exit. The experimental results thus obtained were compared with predictions obtained from the literature and with numerical results obtained from three-dimensional numerical simulations.

Keywords: heat sinks, top inlet, side exit, experimental data, numerical simulation.

1. Introduction

The parallel plates heat sink with impingement cooling constitutes an alternative to remove the electric power dissipated in the microcomputers processors. The increasingly high performance requires more efficient heat sink designs in order to guarantee an operation under a reliable temperature range.

One of the first investigations of parallel plates heat sinks with top inlet using air in the laminar and turbulent regimes was the work of Sparrow, Stryker and Altemani (1985). The presented results were a pressure loss coefficient and also local and average Nusselt as functions of the airflow and the parallel plates spacing. The data were obtained through the naphthalene sublimation to the air and the heat and mass transfer analogy, to simulate the convective heat transfer from two square isothermal plates 0,10 m on a side and four spacings (1.2 – 1.5 – 1.9 and 2.4 mm) between the plates. The tests were performed both without any flow obstruction and with either partial inlet or outlet obstruction.

From a numerical analysis and experimental data, Biber (1977) presented two dimensionless correlations, for a pressure loss coefficient and for an average Nusselt number, for a heat sink with variable top obstruction. These correlations are dependent on the airflow rate and heat sink parameters such as the fins height and length, the fins spacing and the inlet obstruction.

The experimental data of Sparrow, Stryker and Altemani (1985) and the correlations of Biber (1997) were compared with results of numerical simulations performed by Souza and Altemani (2003). The numerical results indicated a much better agreement with the experimental data than with the correlations.

Holahan, Kang, and Bar-Cohen (1999) performed measurements and numerical simulations for the airflow in one channel of a parallel plates heat sink with top inlet and side exit. They presented comparative results for the pressure and the temperature profiles for parallel plates 1 mm apart, 50 mm high and 100 mm long. They did not present results for the pressure drop or the average Nusselt number associated to the flow.

The present work was developed with the purpose to obtain reliable experimental data for three actual heat sinks and not only a single channel between parallel plates. In addition, the spacings between the parallel fins of the heat sinks used in this work were larger than the typical value of 1 mm found in the previous works. The obtained experimental results were compared with the correlations of Biber (1977) and with results from numerical simulations performed with the assumption of flow and heat transfer symmetry in each heat sink channel.

2. Experimental Apparatus

The experimental data were obtained in laboratory using the apparatus indicated in Figure 1. It contained a wooden plenum with two compartments separated by a wall where a nozzle was assembled to measure the airflow rate.

During the tests, ambient air was forced in suction mode, in an open circuit, by a fan located downstream, outside the laboratory room. It passed initially through the heat sink, the nozzle, and then a connecting pipe with a flow-regulating valve before being discharged to the fan and to the ambient outside the laboratory.

A heat sink was positioned at the first plenum compartment so that the forced airflow was directed normal to its base. The air inlet was a square section with an opening of 0.05 m made in a Plexiglas plate. This plate was a 0.20 m square plate with a thickness of 0.015 m. It was mounted to the top of the plenum with eight screws and nuts, sealed by means of a rubber ring, as indicated in Figure 2. Four threaded rods were attached to the Plexiglas plate in order to support the heat sinks and the heaters inside the plenum.

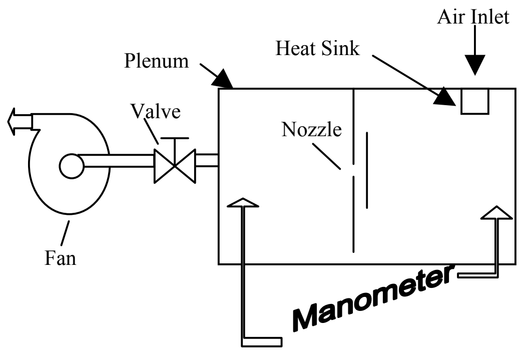

Figure 1. Experimental Apparatus.

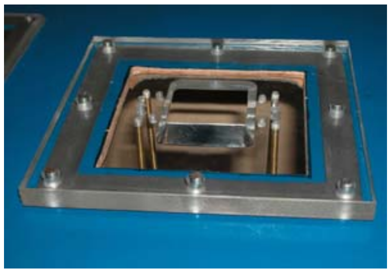

Figure 2. Entrance Plate.

The heating was accomplished by a main heater and a guard heater. The main heater, located just below and in good thermal contact with the heat sink base, consisted of an aluminum block with a 0.05 m square base 0.02 m thick housing five electric resistors inside. The guard heater was located below the main heater, consisting of another aluminum block with similar dimensions housing three other electric resistors. The guard heater was separated from the main heater by a 0.005 m wooden layer. During the tests, both heaters were set to the same temperature, in order to minimize heat losses from the main heater. Each heater was connected to an independent DC power supply. The power dissipation rates were measured by a digital multimeter, reading the DC current and tension in each circuit.

The airflow rate was obtained from the pressure drop through the nozzle, using an inclined manometer fulfilled with ethylic alcohol. The atmospheric pressure and the ambient temperature were also measured during each test.

The temperatures were obtained by ten thermocouples located as follows: two in the heat sink base, two in the main heater, two in the guard heater, two in the Plexiglas plate and two around the heat sink inside the plenum. The thermocouples were of the J type (iron-constantan) with coated 0.25 mm diameter wires. They were obtained by an electric arc under an ambient with Argon to avoid oxidation. The reference temperature was obtained by an equilibrium bath of ice and liquid water.

For the experimental tests, three heat sinks were built using aluminum for both the base and the fins. Each heat sink base was an aluminum block with a square section 0.05 m on a side and a thickness of 0.01 m. The fins were cut from a 1 mm plate and fitted into 3 mm deep grooves machined in each base, as indicated in Figure 3. The fins fitted snugly into the grooves and in order to increase the thermal contact, a mixture of epoxy glue with zinc oxide was used to assemble the heat sinks. The three heat sinks consisted respectively of equally spaced 15 fins 15 mm tall, 11 fins 25 mm tall, and 8 fins 50 mm tall, as indicated in Figures 4 to 6.

Figure 3. Heat Sink Base.

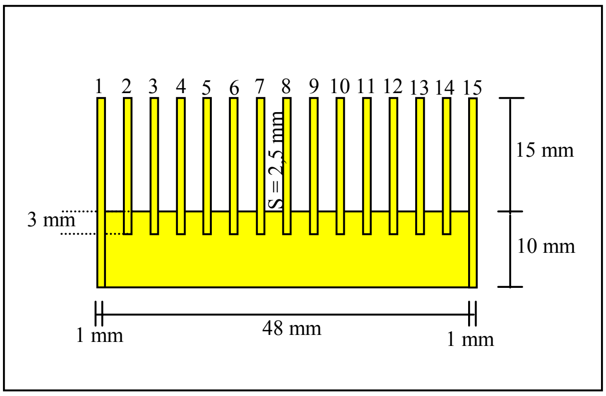

Figure 4. Heat Sink I.

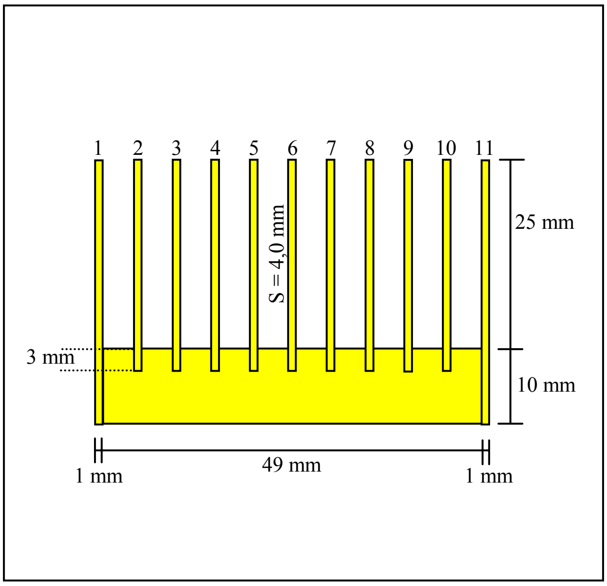

Figure 5. Heat Sink II.

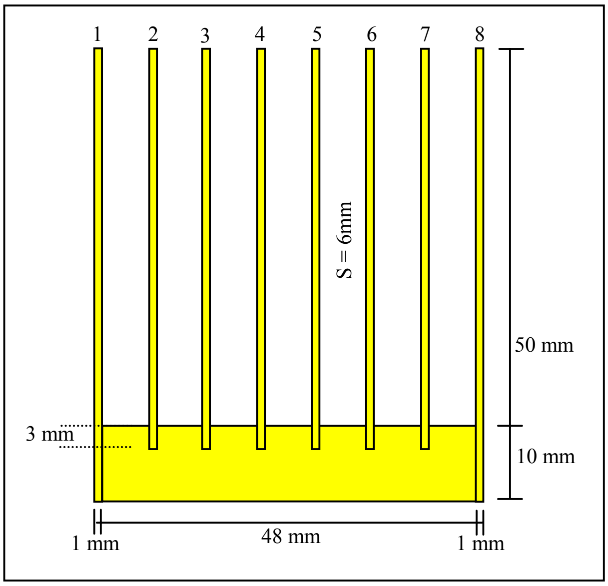

Figure 6. Heat Sink III.

3. Experimental Procedure

The flow path through the heat sink with top inlet and side exit is indicated in Figure 7 – this is known as the TISE configuration. The total pressure drop was measured as the difference between the laboratory atmospheric pressure and that inside the upstream plenum chamber. This procedure was based on the work of Sparrow, Stryker and Altemani (1985).

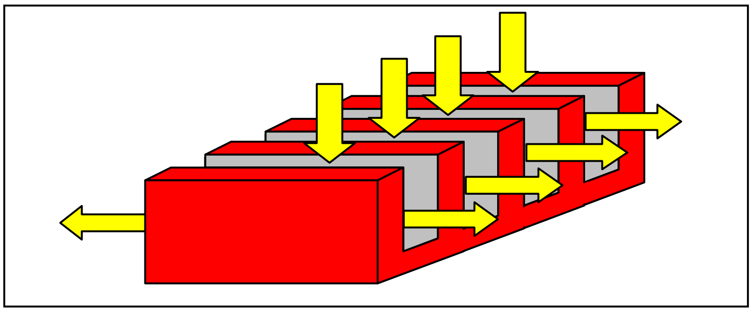

Figure 7. TISE (Top Inlet Side Exit) Configuration.

The thermal tests were performed with the heat sink base temperature around 20 °C above the laboratory ambient temperature. This was obtained in each test varying the power dissipation in the main heater resistors. The convective heat transfer was obtained by indirect means, subtracting the estimated thermal losses from the supplied electric power. The thermal losses considered in this work will be described next.

The sides of the heat sink and the main heater were insulated with a rubber material and the corresponding conductive-convective losses to the air inside the plenum were estimated. The heat sink lateral fins were in contact with the Plexiglas plate indicated in Figure 2. Due to this contact, the walls of the entrance plate were modeled as a fin to estimate their contribution to the thermal losses. The radiation losses from the fins and the heat sink base surfaces were also estimated assuming gray body behavior. The thermal losses from the thermocouple wires connected to the heat sink and the main heater, as well as from the electric power wires connected to the main heater were modeled as infinite long fins. The power loss due to ohmic dissipation in the connecting wires from the DC power supply to the main heater was also included in the analysis.

The sum of all the estimated thermal losses was about 10 per cent of the supplied electric power. The largest contribution was that due to the sides of the heat sink and main heater, around 36 per cent of the total losses, and the smallest, that due to the thermocouple and electric wires, corresponding to 4 per cent of the losses.

The experimental data in each test were obtained under steady state conditions, obtained after about three hours. To check this condition, the data were collected each 30 minutes and compared with the previous values.

The uncertainties of the experimental results were estimated as proposed by Kline and McClintock (1954). Actually, the uncertainties estimated for each measured value were input to a facility included in the software EES (Engineering Equation Solver), where their propagation to the results was obtained according to the method described by Taylor and Kuyatt (1994).

4. Numeric Simulation

The conservation equations of mass, momentum and energy associated to the considered heat sink and airflow were solved numerically through the finite volumes method. The solutions were obtained with the software PHOENICS, version 3.5.1, using a microcomputer equipped with a Pentium 4 with 2 Ghz processor and 512 MB RAM.

The flow and convective heat transfer were assumed symmetric inside the heat sink channels, so that the calculation domain was reduced to that indicated in Figure 8. It consists essentially of one quarter of a heat sink channel, with one half-fin thickness, and the corresponding dimensions are indicated in Table 1. A total of 34,000 control volumes were distributed non-uniformly in this domain. The convergence control for the numerical solution was established in terms of the global residual value for each conservation equation, limited to 0.1 per cent of the total inflow of mass, momentum or energy. The boundary conditions in the x coordinate direction were those of symmetry at both boundaries. In the y direction, a uniform inlet was assumed at the north boundary while the opposite south boundary contained an outlet. In the z direction, symmetric conditions were assumed at the positive face, and at the negative face three distinct conditions, indicated in Figure 8, were considered involving the lateral entrance, the wall of the entrance plate and the lateral outlet.

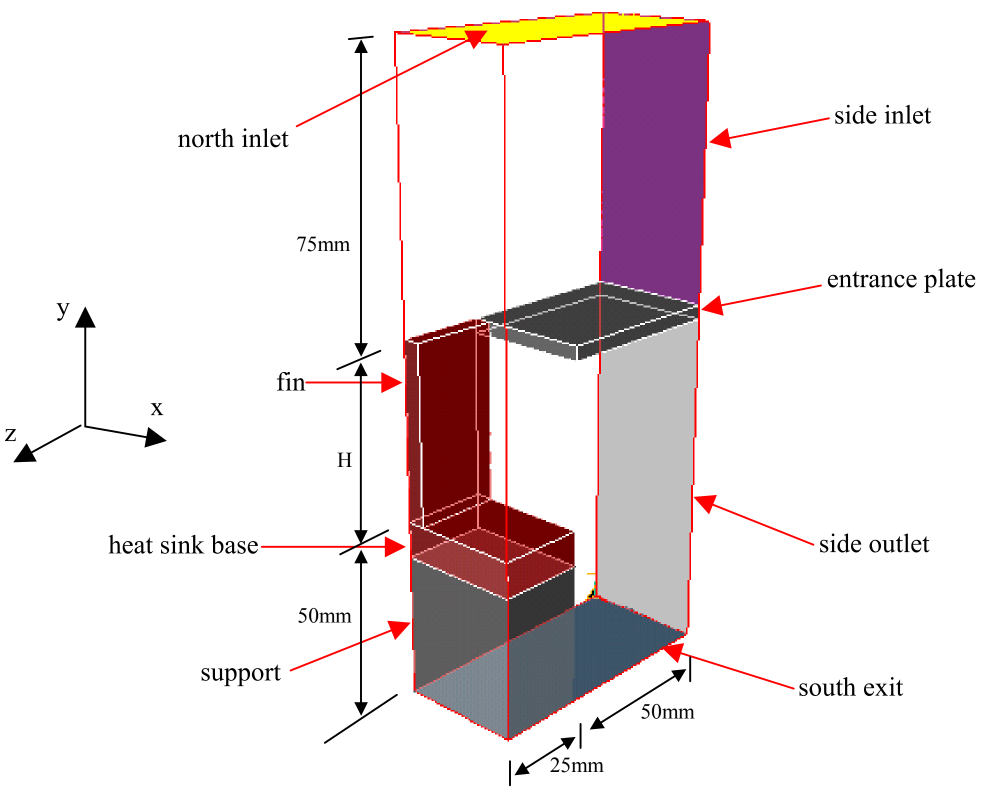

Figure 8. Numerical Domain.

Table 1. Numerical Domain Dimensions.

Description	x (mm)	y (mm)	z (mm)	Observation
Numerical Domain	0,5 + S/2	50 + H + 75	75	S → fins spacing H → fins height
Support	0,5 + S/2	40	25	
Heat sink base	0,5 + S/2	10	25	
Entrance plate	0,5 + S/2	1	50	
Fin	0,5	H	25	

The flow properties were assumed constant, the air density and viscosity evaluated at the entrance temperature and all the remaining properties evaluated at the film temperature between the entrance and the heat sink base. All the results were obtained under steady state conditions, with an average computing time for each test run of about 3 minutes.

Both laminar and turbulent flow regimes were included in the simulations performed. The turbulent flow was treated with the algebraic LVEL model, developed by Agonafer, Liao and Spalding (1993), using Spalding's law of the wall (1961). The turbulent viscosity is evaluated locally in terms of the distance to a wall, the local velocity and the laminar viscosity. This turbulence model is included in the PHOENICS software.

5. Results and Discussion

The experimental results, together with those from the correlations and from the numerical simulations will be presented now, in connection with Figures 9 to 14. All the results were obtained for a range of airflow rate from 0.002 m^3/s to 0,012 m^3/s.

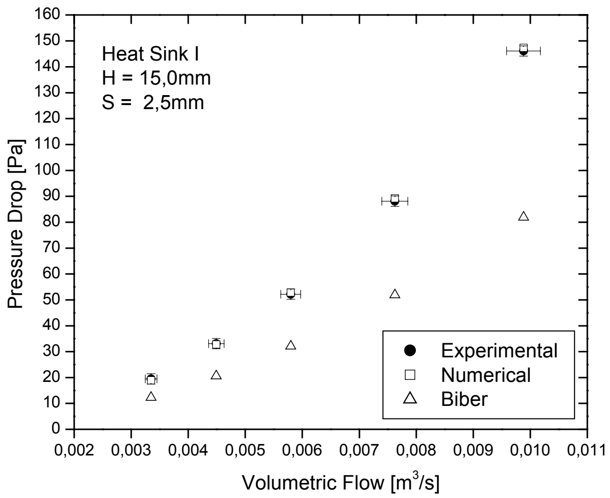

Figure 9. Pressure Drop – Heat Sink I.

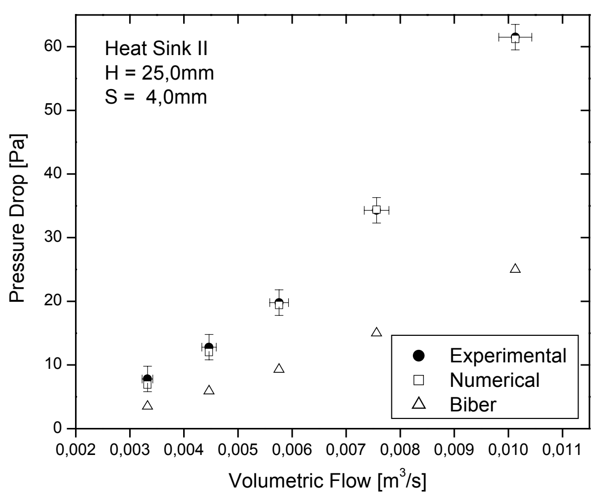

Figure 10. Pressure Drop – Heat Sink II.

For the heat sinks I, II and III the pressure drop predicted by the numerical simulations were within an average deviation respectively equal to 2 %, 5 % and 9 % from the experimental data. The corresponding average deviations for the correlation of Biber (1977) were 40 %, 56 % and 81 % with respect to the measured values.

In general, compared to the experimental data for the three heat sinks, the correlation due to Biber (1977) presented an average deviation of 60 %, while the numerical predictions were within 6 % of the data. The results presented in Figures 9 to 11 indicate that the correlation under predicted the experimental results for all the heat sinks, showing also an increasing deviation with the flow rate.

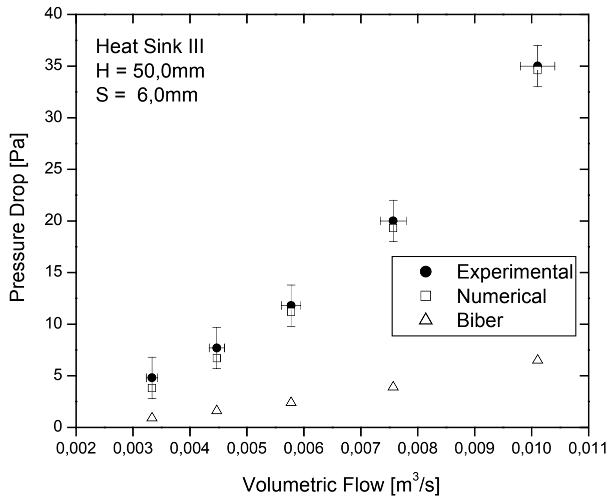

Figure 11. Pressure Drop – Heat Sink III.

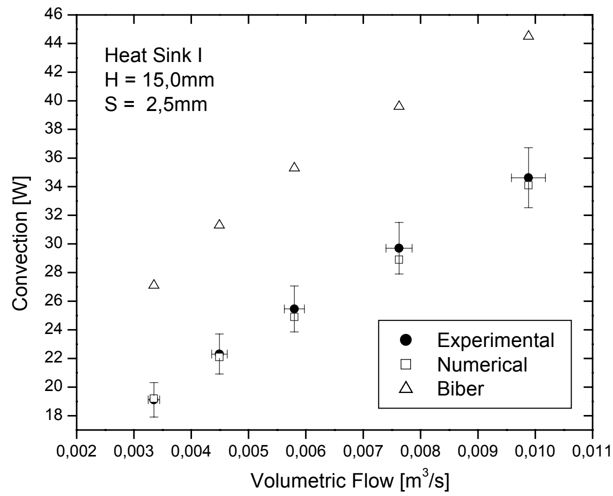

Figure 12. Convection – Heat Sink I.

The results for the convective heat transfer presented similar trends for the three heat sinks, as indicated in Figures 12 to 14. The numerical simulations presented a very close agreement with the experimental data, showing an average deviation of only about 2 %. The predictions due to the correlation of Biber (1977) over predicted the measured values with an almost uniform value about 31 % for all three heat sinks.

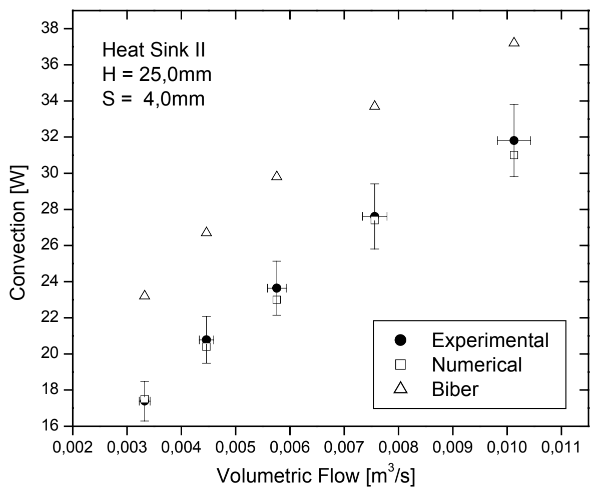

Figure 13. Convection – Heat Sink II.

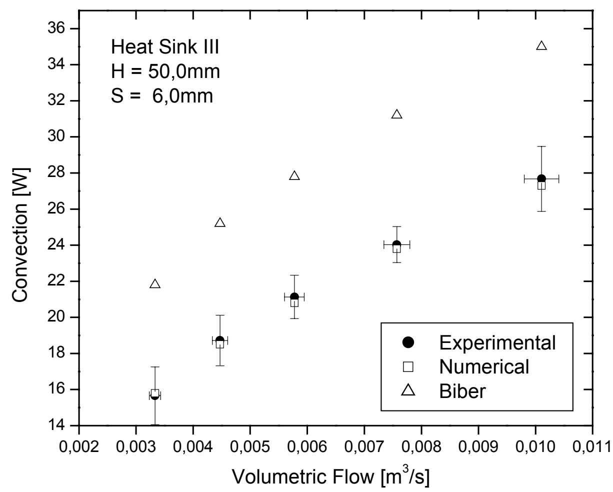

Figure 14. Convection – Heat Sink III.

6. Conclusions

The purpose of this work was to perform an experimental analysis of parallel plates heat sinks with top inlet and side exit. Experimental data for the flow pressure drop and the convective heat transfer were compared with results obtained from numerical simulations and from correlations presented by Biber (1977).

In comparison with the experimental data, the results from the numerical simulations presented an average deviation equal to 6 %, while those from the mentioned correlation under predicted the experimental data and presented an average deviation equal to 60 %.

For the convective heat transfer, the numerical simulations were within 2 % of the experimental data. The correlation presented in Biber (1977) showed an almost uniform over prediction equal to 31 % in comparison with the data measured in this work.

The experimental investigation performed in this work was important to explore and obtain reliable laboratory data with actual heat sinks and not only with a single parallel plates channel.

The numerical simulations indicated, with their results, that they offer an excellent design alternative for heat sinks, due to their flexibility to analyze distinct heat sinks and flow conditions. The confidence obtained with the numerical analysis, through the comparison with the experimental data, some optimization studies are being performed concerning the heat sink design.

7. Acknowledgements

The present work was performed with the support of CNPq (Conselho Nacional de Desenvolvimento Científico e Tecnológico – Brasil) in the form of a Scholarship to the first author.

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