Performance Improvement on Water-cooled Cold-Plate

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Abstract: -With the rapid growth of electronic technology and industry, heat flux generated in a chip of the integrated circuit nowadays becomes so large that reliable operation needs faster cooling rate than ever. Comparatively high thermal capacity in liquid makes it more attractive as a coolant than just to increase the flowing rate of the cooling air. Water definitely has the advantage but also has the disadvantage of non-dielectric property. Thus, a cold plate has to be used to contain the cooling water inside. Channels have been cut within the cold plate in order to increase the flowing passages of water within the limited space defined by the size of the IC chip. Further heat transfer enhancement could be attained by roughing the surface of the channel wall, changing the configuration of the channels in to pin structure for water flow. In the present study, the effects of swirling inflow of the cooling water on the heat transfer have been studied in order to further increase the heat transfer rate of the water cooled cold plate.

Key-Words: -Swirling flow, liquid cooler, mini channel

1. Introduction

As water has much higher thermal capacity and much less specific volume than air, it unavoidably becomes a popular coolant in cooling down the high heat flux generated from the IC chip. In order to effectively cool down the hot surface of the chip, the cold plate made of copper and with cooling water flowing inside it has been applied generally. It could be easily mounted on the hot surface of the IC package without any trouble in solving the short circuit problem by the water flow. To attain adequate cooling rate from the rather small space limited by the size of the chip, the pin studded cold-plate (as shown in Fig. 1) and the mini channel cold-plate are commonly adopted. The design of pin flow cold-plate can effectively increase the contacting opportunity of the cooling water and the cold plate. The pins also disturb the flow field of the cooling water, and hence produce the turbulence in the water flow resulting the increase of heat transfer rate of the cold plate.

Another type of cooling flow inside the cold plate is shown in Fig. 2. Mini-channels are cut into the copper block to form a long winding path in order to give the coolant a maximum contact with the cold plate.

In order to further augment the heat transfer rate of the cold plate with conventional configuration of the cold plate, either increasing the wetted surface area or making the flowing pattern should be tried. Thus in the present study cold plates of three different design were made and measured to learn the effects on their heat transfer rate. A new mini channel design is one of them. With the pin fin cold-plate design, attention has paid on changing the flow pattern and velocity of the coolant. Swirling flow and spray flow are considered to do the job.

These three types of cold-plates investigated in the present study are:

(1) The double spiral channels. It increases the number of the flow channels and hence can take more heat away as shown in Fig. 3.

(2) The swirling flow channel type A. Swirling water flows are formed by the shape of the channel and the water will be accelerated as shown in Fig. 4.

(3) The swirling flow channel type B.
Falling liquid films are formed along the surface of the copper pins before they turn into swirling flow in the channels as shown in Fig. 5.

Comparison with a commercial cold plate is necessary to check the degree of improvement in heat transfer rate. A local brand, the Tai-Chi water-cooling cold-plate as shown in Fig. 2 was chosen for comparison.

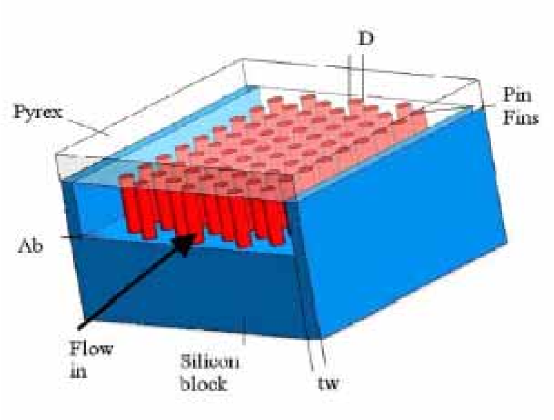

Fig. 1 The pin-studded cold plate

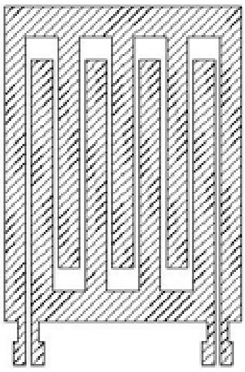

Fig. 2 The mini-channel cold plate

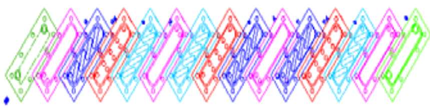

Fig. 3 The double spiral channel cold plate

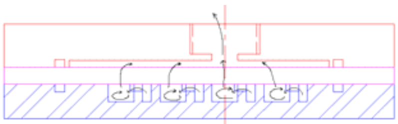

Fig. 4 The swirling flow type A cold plate

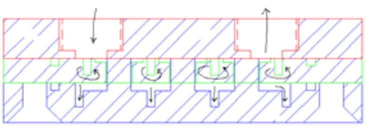

Fig. 5 The swirling flow type B cold plate

2. Some related literature survey

2.1. "Next Generation Devices Electronic Cooling With Heat Rejection to Air" [1]

Conventional technology to cool desktop computers and servers is that of the "direct heat removal" heat sink, which consists of a heat sink/fan mounted on the CPU. Although this is a very cost effective solution, it is nearing its end of life. This is because future higher power CPUs will require a lower R-value than can be provided by this technology, within current size and fan limits. This paper discusses new technology that uses "indirect heat removal" technology, which involves use of a single or two-phase working fluid to transfer heat from the hot source to an ambient heat sink. This technology will support greater heat rejection than is possible with the "direct heat removal" method. Further, it will allow use of higher performance air-cooled ambient heat sinks than are possible with the "direct heat removal" heat sink. A concern of the indirect heat removal technology is the possibility that it may be orientation sensitive. This paper identifies preferred options and discusses the degree to which they are (or not) orientation sensitive. It should be possible to attain an R-value of 0.12 K/W at the balance point on the fan curve.

2.2. "Analysis of Three-dimensional Heat Transfer in Micro-channel Heat Sinks." [5]

In this study, the three-dimensional fluid flow and heat transfer in a rectangular micro-channel heat sink are analyzed numerically using water as the cooling fluid. The heat sink consists of a 1-cm2 silicon wafer. The micro-channels have a width of 57 μm and a depth of 180 μm, and are separated by a 43 μm wall. A numerical code based on the finite difference method and the SIMPLE algorithm is developed to solve the governing equations. The code is carefully validated by comparing the predictions with analytical solutions and available experimental data. For the micro channel heat sink investigated, it is found that the temperature rise along the flow direction in the solid and fluid regions can be approximated as linear. The highest temperature is encountered at the heated base surface of the heat sink immediately above the channel outlet. The heat flux and Nusselt number have much higher values near the channel inlet and vary around the channel periphery, approaching zero in the corners. Flow Reynolds number affects the length of the flow developing region. For a relatively high Reynolds number of 1400, fully developed flow may not be achieved inside the heat sink. Increasing the thermal conductivity of the solid substrate reduces the temperature at the heated base surface of the heat sink, especially near the channel outlet. Although the classical fin analysis method provides a simplified means to modeling heat transfer in micro-channel heat sinks, some key assumptions introduced in the fin method deviate significantly from the real situation, which may compromise the accuracy of this method.

2.3. "Theoretical and experimental investigation on swirl atomizers." [6]

Theoretical and experimental studies have been made to investigate the influence of atomizer construction and controlled pressure difference of swirl atomizers. The analysis of fluid field in the swirl chamber is governed by mass/energy conservation rules; in the region outside the nozzle, the analysis of oscillation of liquid sheet is based on Squire's expression for the amplitude growth rate. With some physical assumptions of control volume, initial values and model correlation, analytical results make it possible to predict film thickness, velocity distribution, spray cone angle and droplet size directly. A series of experiments have been carried out to support and compare the theoretical results. The pictures for velocity profile and boundary layer thickness in the swirl chamber have been established with the aid of MATLAB. Finally, it has been recognized that the change of different individual design parameter and its corresponding flow number would still be a way to optimize the performance of swirl atomizers.

Experimental apparatus and procedures:

The purpose of the experiment is to determine and compare the heat transfer rate of the swirling channel, the double spiral channels, and the traditional design cold-plate.

3. Experimental Setups

The experimental apparatus is shown in Fig. 6. It includes the cold plate to be tested, a hybrid recorder, a constant temperature reservoir, a flow meter, and a CPU simulation heating block.

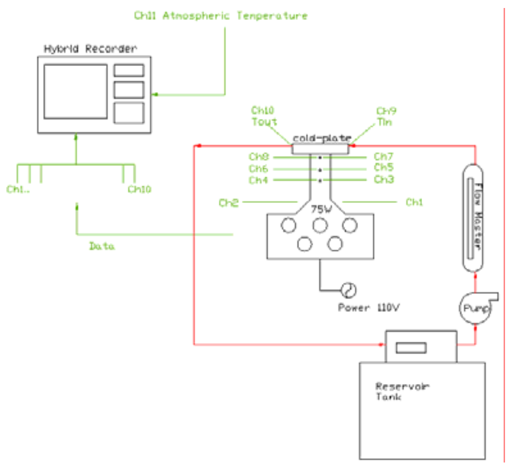

Fig 6 The experimental apparatus

Detail structure of the CPU simulation heating block is shown in Fig. 7. The heat dissipating plane of the cold plate is made of copper. 5 electrical heaters of 15W each are installed at the bottom block. The heat generated is transferred to the cold-plate through the copper rod of 1 cm^2 cross sectional area. Heat flux transferring to the surface of the cold plate is determined by 6 thermocouples embedded at 3 different heights separated 1 cm each in the copper rod. Inlet and outlet temperatures of the cooling water are also measured.

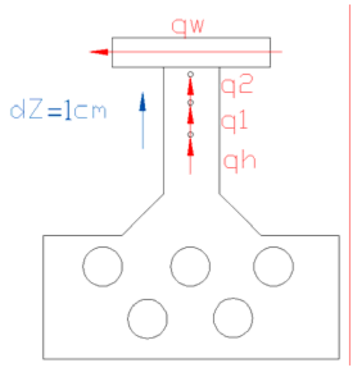

Fig 7 The test section

The heat transfer rate to the bottom surface of the cold plate is determined by 1-D heat conduction through the copper rod. Fourier conduction law gives q = -kAΔT/Δz. q_h is the heat transfer between heat source and the first layer. q_1 is the heat transfer between the first and the second layers, and q_2 is the heat transfer between the second and the upper layer. q_w is the heat transfer rate between the top layer and the water flow. Theoretically, $q_h = q_1 = q_2 = q_w$, but in fact $q_h \leqq q_1 \leqq q_2$. Then q_w can be found from the equation q = UA ΔT_{lm} where U is the total heat transfer coefficient, A is the contact area, and ΔT_{lm} is the log- mean temperature difference defined by:

$$\Delta T_{lm} = \frac{\Delta T_{\max} - \Delta T_{\min}}{\ln \dfrac{\Delta T_{\max}}{\Delta T_{\min}}} .$$

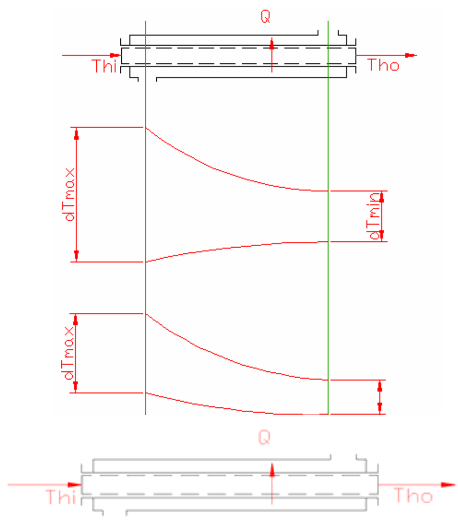

Fig 8 The temperature distributions

As cooling flow passes through a channel shown in Fig. 8, temperature drops exist between both in the inlet-and-outlet water flow and across the channel. Thus the log mean temperature has to be calculated. Since the channel is very narrow, the very small temperature drop across it could be negligible. Thus just the temperature drop between the inlet and outlet water flow has to be considered. Then we can easily find the log-mean temperature ΔT_{lm}. Theoretically the relationships of $q_h = q_1 = q_2 = q_w =$ UA ΔT_{lm} exist. Preliminary consider the influence from the difficult variation of the channel in the cold-plate is contain in U, then we can find overall heat transfer coefficient U for given values of q_w, A, and ΔT_{lm}. Obviously the cold plate with higher U is the one with better performance.

4. Discussions

4.1. Steady state temperature:

The test section is heated by a fixed heat source of 75W power. Cooling water flows through the channel at 0.5 l/hr and at 30°C. Temperature of CPU is analyzed when it reaches steady state. This method is the most general one for the analysis of the water-cooling system. As the gap across the channel to the top surface of the cold plate is very small and so is the temperature drop, the measured temperature could be taken to be the CPU surface temperature. Value of the measured temperatures of channel 7 and 8 versus time are plotted in Figs. 9, 10, and 11 for four different cold plates. From these figures, the steady state temperature for Tai- chi cold plate is 62.4°C, while for the swirling channel type A is 46.9°C; for the swirling channel type B is 48.5°C; and for the double spiral channel is 52°C. The higher the temperature is, the lower the heat transfer rate is.

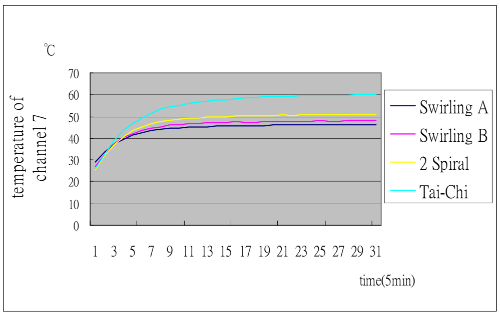

Fig 9 Measured temperature of channel 7

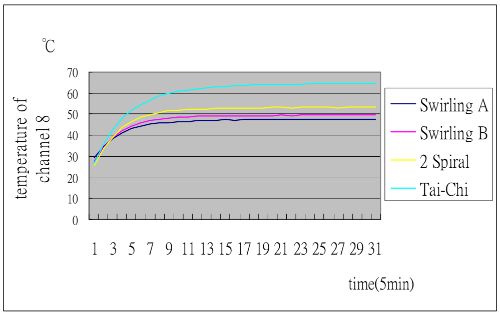

Fig 10 Measured temperature of channel 8

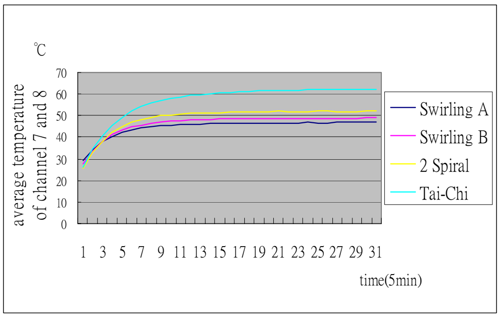

Fig 11 Avg. temperature of channel 7 & 8

4.2. Heat transfer rate by heat conduction:

From q=-kAΔT/Δz, we can find the heat transfer rate between two layers. The relationship of q_1, q_2, versus time are shown in the following diagrams.

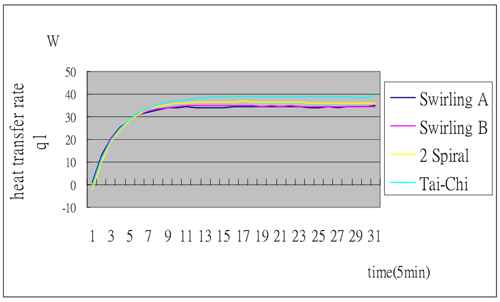

Fig 12 Heat transfer rate by heat conduction

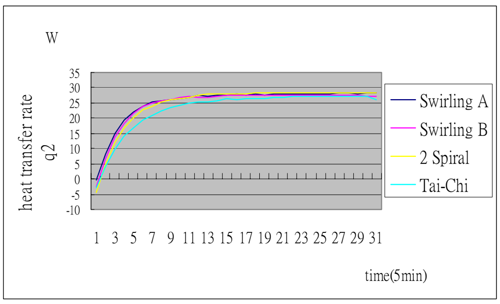

Fig 13 Heat transfer rate q2

4.3. Calculated heat transfer rate from the temperatures at the flow's inlet and outlet:

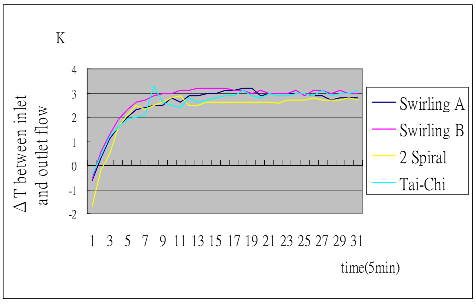

Fig 14 Inlet-outlet temperature drop

The measured inlet-outlet temperature drop is shown in Fig. 14. Fig. 15 shows the relationship between q_W versus time from

$q_w = \dot{m}_w C_p (T_{out} - T_{in})$.

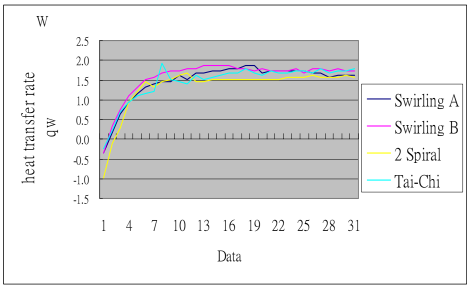

Fig 15 Heat transfer rate versus time

4.4. The logarithmic mean temperature ΔT_{lm}

As flow through a heat exchanging channel shown in Fig. 8, temperature drops exist in both at the inlet and outlet of the water flow and across the channel. Thus the log-mean temperature of the exchanger is determined by the equation and plotted in Fig. 16.

$$\Delta T_{lm} = \frac{\Delta T_{\max} - \Delta T_{\min}}{\ln \dfrac{\Delta T_{\max}}{\Delta T_{\min}}}$$

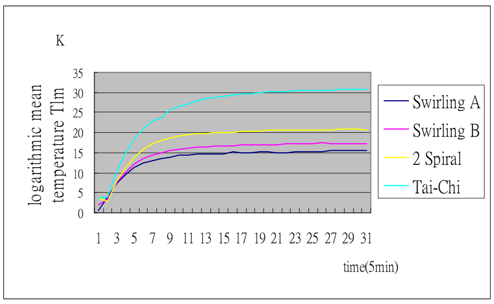

Fig 16 △T_{lm} versus time

4.5. The overall heat transfer coefficient:

The overall heat transfer coefficient of a system has a similar trend with the heat transfer rate and heat transfer effectiveness.

As U= $\dfrac{q}{A\Delta T_{lm}}$ from q=UA ΔT_{lm} .

Then we have three given q values q_1 , q_2 and q_w . Theoretically, the relationship $q_1 = q_2 = q_w$ =UA ΔT_{lm} exist. If the thermal insulation is not adequate, some heat might lose to the environment and q_1 , q_2 , and q_w will not be equal. A reasonable value is needed for calculation. Finding it from solid conduction is more reliable. q_2 has been chosen as its position is closer to the CPU surface. The calculated values for U are: 9000 W/m^{2}K for the swirling channel type A cold plate; 8000 W/m^{2}K for the swirling channel type B cold plate; 4600 W/m^{2}K for the double spiral channel cold plate; and 1900 W/ m^{2}K for the Tai-Chi cooler. The results are plotted in Fig. 17. The improvement in heat transfer rate obtained by increasing the channel length and forming swirling flow is very significant.

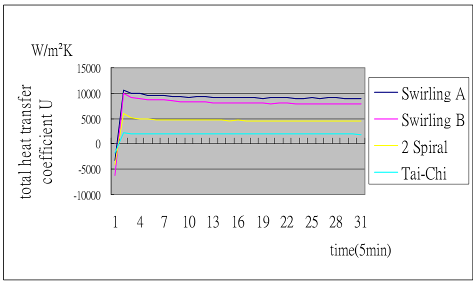

Fig 17 Overall heat transfer coefficient

5. Conclusions

Several conclusions have been obtained from the results of this experimental study.

(1) Contacting area of Tai-Chi water-cooler and the double spiral channel are about the same, but we can use the simple flow channel design to increase the contacting time duration of water in the cold-plate and obtains greater heat transfer rate.

(2) Generally, a higher flow velocity yields a higher heat transfer rate. But higher flow velocity also shortens the contact time of cooling water with the cold plate. Water will be accelerated in the cavity of the cold-plate with swirling channel. Even though the water is accelerated, however, due to the swirling motion in the cavity, the contacting time of the water and the cold plate could also increases.

(3) The pin structure for cooling flow to cross and the square flow channel of Tai-Chi have

similar disadvantage. The inlet and outlet flow are not divided. The outlet flow may cause backflow and decrease the heat transfer rate.

(4) As the water flow in the swirling channel cold-plate type *A* goes through a narrow gap and into the flow-collecting cavity, spray of water may be formed to cause phase change and then take more heat away.

(5) The design of double-spiral cold plate increases the length of channel flow. Swirling flow channel type A cold plate focus on the effect of flow swirling. Swirling motion can produce a faster cooling flow velocity. Copper pins have been put into every swirling cavity to increase the contacting surface area in the swirling flow channel type B. Consequently they also reduce the swirling motion of the flow. Thus the overall heat transfer ability is decreased. In comparison, the swirling flow channel type A cold plate has the best performance. Swirling flow channel type B cold plate has better performance than the double-spiral cold plate. Flow velocity is the most essential factor. Adding pins are more influential than increasing the length of the cooling flow channel.

(6) Further studies can be focused on the formation of liquid droplets in addition to the swirling motion of the cooling flow. Small droplets are easier to be vaporized to carry away the latent heat. With limited pressure drop supplied by a small size pump, it is a challenge to obtain a fine droplets distribution for high heat dissipation flux.

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