Heat-dissipative performance of heat sink in desktop PC

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Abstract

This paper presents different test conditions intended to investigate the factor of heat-dissipative performance of heat sinks in desktop PC. For this study, three test factors, such as thermal-interface materials (TIMs), static load between heat sink and central processing unit (CPU), and fan speed have been selected. Experimental results indicate that TIMs have a major impact on the heat-dissipative performance; the effective temperature-reducing ratio of the CPU simulator was reached by as much as 43.8%. From experimental results, we conclude that the most efficient solution for the CPU heat problem is to simplify the attachment mechanism between the CPU and heat sink so as to ensure the effectiveness of the TIMs between their surfaces. This solution is simpler and more economical than improving heat the sink fins.

Keywords: Heat sink, Thermal-interface materials, Static load, Fan speed

1. Introduction

Increased operating frequencies and smaller packaging sizes have enabled the development of increasingly powerful personal computers (PCs). However, the inevitable increases in temperature in the central processing unit (CPU) pose the engineering problem of developing adequately active cooling devices. Therefore, development of appropriate heat-dissipation CPUs has become a critical problem. The effective dissipated heat of the CPU is an important performance measure for cooling devices that ensure that the CPU is operating at a safe temperature. Temperatures exceeding the maximum operating limit of a CPU may cause irreversible damage to the thermophysical properties of this CPU. A CPU-cooling device is therefore essential for maintaining the temperature of the CPU. The major parameters involved are the contact pressure between the heat sink and the CPU, the thermal-interface materials, and the fan speed.

The most common active-cooling strategy in the context of desktop PCs is the dissipation of heat from the extended surface of the heat sink by the forced airflow produced by a fan. Consequently, because heat is transferred across the heat sink–CPU interface through surface-asperity microcontacts and air-filled microgaps, thermal contact resistance is a major determinant of heat transfer. Traditional coolers with extruded or bent fins located on a thick aluminum or copper base become inefficient even at heat loads of 70–100 W [1,2]. The mean value of the total thermal contact resistance is typically 0.3–0.5 °C/W [1,2]. The other drawbacks are their considerable masses and the high noise level (up to 30–40 dB) produced by the high-speed fans. Of the many cooling technologies developed recently, heat pipes [3–9] and vapor chambers [10] are the most promising. The main concept of the cooling approach is to reduce the total thermal resistance to 0.17–0.22 °C/W by increasing the efficiency of heat transfer from the CPU–contact interface to the heat sink. This approach requires considerably smaller heat-sink fins and enables the use of high-speed fans. However, the large mass and size parameters of the
cooler are problematic. Pastukhov and Maydanik [11] developed an active CPU cooler with copper–water loop heat pipes for desktop PCs. Their experimental results indicated that the maximum heat-transfer capacity of the cooler was 500~600 W. Minimum values of the total thermal resistance of the cooler were 0.15 and 0.17 °C/W at heat loads of 500 and 250 W, respectively. Another experimental study of a heat-sink assembly with oblique straight fins by Lin et al. [12] found that, for a CPU with 82 W power consumption, a high-pressure fan operating at 2000 rpm reduced the temperature of the CPU case by 6 °C. The study also showed that the static pressure of the cooling fan is the dominant factor in the effectiveness of a heat-sink assembly.

Another factor that affects thermal contact resistance is deformation in the amplitude of the contact interface under a static load. For a given contact interface, the asperity deformation is greater for a larger static load than for a smaller one. Therefore, the effective contact–interface area is larger, and the thermal contact resistance is lower. Static load is positively related to asperity deformation but negatively related to thermal contact resistance. The most common method of reducing thermal contact resistance is to introduce a thin layer of thermal interface materials (TIMs) between the contact interfaces [13–15]. Most studies of TIM in literature have focused on the effects of thermal conductivity on thermal contact resistance [16–20]. Notably, various TIMs suitable for thermal enhancement in electronic systems have been investigated. The results of these studies are useful when a TIM is applied for enhancement of the thermal contact conductance of a CPU–heat sink assembly. The thermophysical properties of the TIM may also affect the thermal contact resistance. High values of thermal conductivity and thermal expansion coefficient have a favorable effect on thermal contact resistance [21]. For traditional TIMs, high compliance is due to the fluidity of the base materials, such as silicone oil, whereas high conductivity results from the filling particles, such as silver powder. To achieve improved performance, several research works have been conducted in the search for better filling materials and optimum volume-fraction ratio [22–24]. However, with long-term use of TIMs, the gel-like components dry out with time and their thermophysical properties change. However, most TIMs are difficult to apply due to their limited thermophysical properties and poor adherence to surfaces, which degrades thermal contact conductance.

This study experimentally analyzed the heat-dissipative performance of the heat sink of a desktop PC. The objective was to obtain experimental data that could be used to improve the heat-dissipative performance of heat sinks. Because the heat sink is clamped to the CPU by an attachment mechanism, the structural design used a high level of clip stiffness that resists local board curvature under the heat sink, which is designed to maintain a uniform contact surface between the simulated CPU and the heat sink. Static load was analyzed using a compression/tensile tester between the simulated CPU and heat sink. The effects of static load, TIMs, and fan speed on the thermal dissipative performance could then be determined.

2. Experimental apparatus and methods

2.1 Experimental setup

Figure 1 schematically depicts a diagram of the experimental apparatus. The facility included an attachment mechanism, a CPU-simulator system, and a cooling system. The attachment mechanism consisted of a fixture, a digital force gauge, and a manual type compression/tensile tester that could measure static loads from 8 kgf to 20 kgf with ±0.5% accuracy. The major functions of the attachment mechanism were to apply static load to the base of the heat sink to maintain the desired pressure on the TIM. The CPU-simulator system consisted of three
specimens: a copper plate (33×33×2.9 mm, k=401 W/mK), an electrical heater (33×33×1 mm), and a bakelite base (k=1.4 W/mK). The copper plate and the electric heater were positioned using a fixture of the attachment mechanism on the bakelite base with four fasteners.

This study measured the heat-dissipative performance in a simulated Intel Pentium IV processor. The top surface of the CPU-simulator system was designed to interface with the heat sink of the cooling system. The experimental thermal solution was an active-cooling design with a fan installed at the top of the heat sink. The apparatus contained no other components that might generate heat.

The heat generated by the CPU-simulator system within a chassis (245×265×510 mm) of the experimental apparatus had to be removed to provide an adequate operating environment. Heat generated by the CPU simulator was dissipated by maintaining airflow from the external ambient environment through the experimental apparatus. The chassis design determined the rationality of the experimental results and the precision of acquired data.

2.2 Testing techniques

The experimental tests were conducted in normal environmental conditions at an ambient temperature of 23±1.5 °C. Table 1 shows the contact interface of the CPU simulator and the heat sink with and without the TIM that was selected for each parameter (static load F and fan speed N). An electrical heater was used to simulate the heat generated by the CPU. The cooling system was tested on a simulator with a 100 W capacity. The simulator was insulated from external heat. Three temperatures were measured by the type-T, AWG30 thermocouples. One thermocouple was set at the contact interface of the CPU simulator, and another was set at the other side of the copper core at a distance of 3.5 mm. These thermocouples were secured with high-conductivity thermal grease (k = 4.8 W/mK). The simulator temperature was measured at the geometric center on the contact surface of the copper plate. To ensure an accurate temperature measurement, special care was taken when arranging the
thermocouple bead at the end of a milling groove near the top surface of the copper plate. Then, a thermocouple wire was placed inside the groove and filled with adhesive. A thermocouple was also placed 8 mm above the fan to monitor the ambient temperature. Data acquisitions for all thermocouples were carried out simultaneously at a frequency of 500 Hz.

<table>
<thead>
<tr>
<th>No.</th>
<th>Contact interface of heat sink/CPU</th>
<th>Static load, F (kgf)</th>
<th>Fan speed, N (rpm)</th>
</tr>
</thead>
<tbody>
<tr>
<td>#1</td>
<td>Without TIM</td>
<td>18</td>
<td>2200</td>
</tr>
<tr>
<td>#2</td>
<td>Without TIM</td>
<td>18</td>
<td>2400</td>
</tr>
<tr>
<td>#3</td>
<td>Without TIM</td>
<td>18</td>
<td>2600</td>
</tr>
<tr>
<td>#4</td>
<td>With TIM</td>
<td>12</td>
<td>2200</td>
</tr>
<tr>
<td>#5</td>
<td>With TIM</td>
<td>18</td>
<td>2200</td>
</tr>
<tr>
<td>#6</td>
<td>With TIM</td>
<td>18</td>
<td>2400</td>
</tr>
<tr>
<td>#7</td>
<td>With TIM</td>
<td>12</td>
<td>2600</td>
</tr>
<tr>
<td>#8</td>
<td>With TIM</td>
<td>18</td>
<td>2600</td>
</tr>
<tr>
<td>Intel</td>
<td>With TIM</td>
<td>8.16~31.75</td>
<td>2400~3600</td>
</tr>
</tbody>
</table>

The CPU simulator was characterized by increasing the heat load step-by-step from 15.5 to 87.6 W and with a time lag in the stationary regimes. The maximum heat load was limited by an admissible simulator temperature of approximately 80 °C. The experimental thermal solution was Intel radial-curved-bifurcated fin heat sink (RCBFH; diameter 89.4 mm and height 40.6 mm). The fins and base were made of aluminum, and the core was made of copper. The specifications of the fan of the cooling system were as follows: rated voltage 12 VDC, dimensions L80×W80×H25 mm, and rating speed 3500 rpm. The fan was tested at three speeds (2200, 2400, and 2600 rpm) under the rating speed.

The major tasks in this study were to determine the temperature characteristics and the thermal contact resistances of the cooling system for varying heat loads and other conditions. For the condition of a contact interface without a TIM, only the effects of the static load F and the fan speed N were studied in the range of heat loads less than 61 W. The complete characterization in the presence of TIM, conducted with heat loads up to 87.6 W, tested all possible combinations of the input parameters (totaling eight in number). Eight series of experiments were undertaken for different combinations of TIM, static load, and fan speed (Table 1).

2.3 Theoretical background

To quantify the effect of TIM, static load, and fan speed on heat dissipation in a desktop PC, a theoretical model for thermal management was developed. The thermal profile parameter defined the maximum case temperature (MCT) as a function of the processor-power dissipation, as described in the Intel reference design [25]. The thermal design power and MCT were defined as the maximum values of the thermal profile. The thermal solutions were intended to meet the thermal profile for all system-operating conditions and processor-power levels. The performance of the thermal solution was expressed by the slope on the thermal profile, which represented the thermal contact resistance of the heat sink attached to the CPU.
The thermal contact resistance of the heat sink–CPU assembly, which is the conventional measure of heat dissipation in a CPU-cooling system, was defined as the ratio of the temperature difference between the heat sink and the CPU to the total heat load.

\[
R_C = \frac{\Delta T_C}{Q}
\]  

(1)

where \( \Delta T_C \) is the temperature difference between heat sink and CPU and \( Q \) is the heat load of the CPU simulator. The term \( R_C \) is commonly presented in units of °C/W. The equation for \( R_C \) shows that contact resistance approaches the minimum as \( \Delta T_C \) approaches zero.

Power dissipation and temperature were measured to validate the thermal solutions. Here, the performance of a thermal solution was defined by the thermal characterization parameter \( \psi \).

\[
\psi = \frac{T_{c1} - T_A}{Q}
\]

(2)

where \( T_{c1} \) is the surface temperature of the CPU simulator, \( T_A \) is the ambient temperature above the fan, and \( Q \) is the heat load of the CPU simulator, which is the product of the electric current \( I \) and the electric voltage \( V \) produced by the electrical heater in this study. Unlike thermal contact resistance, the thermal characterization parameter is calculated based on the total CPU power. The three essential measurement parameters for characterizing thermal properties were surface temperature of the CPU simulator, ambient temperature, and heat load of the CPU simulator.

Heat removal from the CPU simulator occurs by conduction across the interface of the CPU simulator surface, through a TIM, into the heat sink, and then by convection to the environment. In the condition of contact interface with the TIM, the heat flux through the heat sink can be calculated using the expression

\[
q^* = -k \frac{T_{c2} - T_{c1}}{\Delta z}
\]

(3)

where \( k \) is the thermal conductivity of the copper core of the heat sink; in this study, the value of \( k \) for copper is 401 W/mK. The value \( T_{c2} \) is the temperature at the end of the copper core of the heat sink. The value \( \Delta z \) is the distance between the two ends of the copper core; in this study, \( \Delta z = 3.5 \) mm.

2.3 Uncertainty analysis

The experiment designed to measure the heat-dissipative performance of the desktop PC may be subject to errors in instrumentation, methodology, and procedure. Experimental uncertainties must be quantified to assess the confidence in the results [26]. The relative uncertainty in the thermal characterization parameter [26] is given by

\[
\frac{\delta \psi}{\psi} = \left( \frac{\delta(T_{c1} - T_A)}{T_{c1} - T_A} \right)^2 + \left( \frac{\delta Q}{Q} \right)^2 \right)^{1/2}
\]

(4)
where $\delta\psi$ is the uncertainty in the result, and $\delta(T_{C1} - T_A)$ and $\delta Q$ are the uncertainties in the variables $T_{C1} - T_A$ and $Q$, respectively. In this equation, $\delta\psi / \psi$ is the relative uncertainty in the result; and the factors $\delta (T_{C1} - T_A) / (T_{C1} + T_A)$ and $\delta Q / Q$ are the relative uncertainties of each variable.

The uncertainties in the surface temperature $T_{C1}$ of the CPU simulator and the ambient temperature $T_A$ above the fan are obtainable similarly. Thus, in this experimental study, the maximum relative uncertainties in the temperatures, the heat load $Q$, and the thermal characterization parameter in the experiments are estimated to be 6.52%, 5%, and 8.22%, respectively.

3. Results and discussion

Three thermocouples were used to monitor the temperature history and to obtain regularized sequential trends that represent thermal contact resistance and thermal characterization parameters.

3.1 The effects of TIM

To reduce thermal contact resistance between the heat sink and the CPU interface, a TIM was used to eliminate gaps in the contact interface by conforming to the mating surfaces. Providing a direct conduction path from the CPU to the heat sink and selecting a TIM with higher thermal conductivity typically improves the heat-dissipative performance. Specifically, the quality of the contact interface between the CPU and the heat sink has a higher impact on the overall performance as the CPU-cooling requirements become stricter. The thermal contact resistance varies considerably depending on the interface geometry, the thermal and mechanical properties of the contact materials, and the interstitial fluid. Surface characteristics such as flatness, waviness, and roughness have major effects on thermal contact resistance. The most common method of minimizing thermal contact resistance is to fill the interfacial gap with a TIM.

This experimental study considered only two limiting cases, the contact interface either with or without a TIM. The thermal grease applied in this study had a thermal conductivity of $k=4.8$ W/mK. This TIM can increase the adhesion and reduce the thermal contact resistance between the bead of the thermocouple and the MCT. A positive approach toward enhancing the heat dissipation of a cooling system is to polish the bottom of the heat sink to reduce surface roughness. This reduces the formation of air gaps. A passive approach is to increase the pressure on the surface of the contact interface to reduce the air gap or to simply coat the bottom of the heat sink with a TIM. This analysis first explored the impact of the TIMs. Figure 2 shows the changes in the measured temperature of the CPU simulator (vertical coordinate) with the heat load $Q$ (horizontal coordinate) at fan speeds of 2200, 2400, and 2600 rpm, when the CPU and the cooling system were integrated under a static load, both with and without TIM.
The goal of this study was to provide an understanding of the impacts of heat sinks for improving the cooling performance imposed on a single processor of Intel Pentium IV. The physical dimensions and the mechanical and thermal specifications of the CPU-simulator system that were used in this study were as specified by the Intel thermal design guidelines [25]. Experimental data for the case of uncoated TIM (Experiments #1, #2, and #3) were compared with those provided by Intel [25]. In the low heat-load condition (Q<26.9 W), the heat generated by the CPU was dissipated by the heat sink. Therefore, the experimental data were lower than the data obtained by Intel. However, as the heat load increased, the thermal contact resistance began to significantly disrupt the heat dissipation, and the discrepancies between these data and the Intel data gradually increased. Immediately after the heat load of the CPU simulator reached 39.2 W, the temperature of the CPU simulator crossed 60 °C under the three different fan speeds, whereas the corresponding temperature reported by Intel was only 53.3 °C. As faster fan speeds reduce the temperature of the CPU simulator, such speeds achieve better heat dissipation. These data show the important effect of forced convection.

In the next experiment, the three fan speeds were set the same as before and the static load was maintained at 18 kgf. The only difference was that the contact surface of the bottom of the CPU simulator and the heat sink were coated with TIMs to improve thermal conductivity. The purpose was to fill in the gaps between the contact surfaces to reduce the increase in temperature caused by the thermal contact resistance. Figure 2 shows the experimental results. The overall findings of this experiment clearly indicate that, after being coated with the TIM, the corresponding heat load of the CPU simulator was significantly reduced. This phenomenon implies that the TIM can weaken the contact resistance, which improves heat transfer from the CPU through the interface and to the heat sink. In addition, it not only reduces temperature of the CPU simulator effectively, but also upgrades the rate of heat dissipation to a higher range. Furthermore, Figure 2 shows that, although the temperatures of the CPU simulator at the three different fan speeds were very close, some differences still existed: the faster the speed of the fan, the lower was the simulator temperature. This confirms the improved heat dissipation achieved by faster fan speeds.

Analysis of the data distribution in Figure 2 shows a significant difference in the availability of coated TIM, especially in the case of higher efficiency of heat-dissipative performance. A comparison of the quantitative data indicated that although the thermal conductivity coefficient of the TIM (k=4.8 W/mK) is much lower than that of
the copper core (401 W/mK), the coating on the heat sink and CPU simulator still weaken the contact resistance significantly. Under the following conditions: heat-dissipative rate \( Q = 61 \text{ W} \), static load \( F = 18 \text{ kgf} \), and fan speeds of 2200, 2400, and 2600 rpm, the effective temperature-reducing ratio of the CPU simulator reached 35.2%, 38.8%, and 43.8%, respectively.

### 3.2 The effects of static load

A mechanism was designed to attach the heat sink directly to the CPU. The attachment mechanism for the dissipation of heat from a CPU, developed to support the CPU-simulator system and the cooling system, was designed to apply a static load on the CPU. A static load ensures that the thermal performance of the TIM between the heat sink and the CPU remains within the minimum/maximum range specified in the CPU. Because of the asperity deformation under the static load, the number of contact spots increases, which in turn decreases thermal contact resistance. Generally, an optimum static load can be found for the thermal contact conductance at the CPU–heat sink interface. Indeed, an excessively low static load causes poor contact, and an extremely high static load causes damage to the CPU–heat sink assembly.

Furthermore, the four groups in Figure 3 represent the \( T_{C1} \) value of the CPU simulator under the conditions of 12 and 18 kgf static load (#7 and #8). The data distribution in Figure 3 indicates that, although the temperatures of the CPU simulator in all the conditions were very similar, the larger the static load (#5 and #8), the lower was the \( T_{C1} \) value of the CPU simulator. This occurs because the static load increases the pressure on the contact surfaces, which then reduces the air gap on its surface, improves thermal contact resistance, and reduces the \( T_{C1} \) value of the CPU simulator.

![Figure 3. Heat-load dependences of CPU-simulator temperatures on static load and fan speed](image-url)

As the thermal characterization parameter \( \Psi \) reflects the heat-dissipative performance, we transformed the experimental data into the respective parameters and plotted them versus the heat load \( Q \) in the diagram, as shown in Figure 4. From the definitions described by Equations 1 and 2, we know that the thermal characterization parameter \( \Psi \) and the thermal contact resistance \( R_C \) have the same unit. In physical terms, the thermal characterization parameter implies the thermal resistance and essentially has an increasing trend following an increased rate of heat dissipation; under conditions of larger static load (#5 and #8), the thermal characterization...
parameter $\Psi$ is slightly lower. Although the variation in the thermal characterization parameter is minimal, the effect of the static load is still distinguishable. This implies that at fixed fan speeds, a larger static load represents a relatively lower thermal resistance. In quantitative terms, when the rate of heat load $Q$ increases from 15.5 W to 87.6 W, the thermal characteristic parameter increases up to 2.2 and 2.3 times in two of the limiting cases (#5 and #8, respectively).

![Figure 4](image)

**Figure 4.** Dependences of the thermal characterization parameter on the heat load and fan speed

The next experiment examined conductive the heat transfer by the heat sink of the cooling system. The distribution of four sets of experimental data in Figure 5 obviously shows that a larger static load implies more conductive heat fluxes through the heat sink.

![Figure 5](image)

**Fig. 5.** Heat-load dependences of the heat flux from the CPU simulator for different fan speeds

### 3.3 Effects of fan speed

The examined cooling system consisted of a heat sink with radial curved bifurcated fin and an axial-flow fan.
The primary function of the fan was to enhance transfer of the heat from the CPU to the environment, which provided efficient heat transfer away from the CPU to a mounted cooling device. Convective heat transfer occurs between the airflow and the surface exposed to the flow and is characterized by (1) the local air temperature above the fan, $T_A$, and (2) the local air velocity across the surface; thus, the cooler the air, the more efficient is the resulting cooling.

The effects of fan speed on the heat-dissipative performance were then examined at fan speeds of 2200 rpm (#4 and #5), 2400 rpm (#6), and 2600 rpm (#7 and #8). Table 1 shows the test results. However, the Intel cooling system implements a variable speed fan. The required fan speed necessary to meet thermal specifications is controlled by the fan inlet temperature. Figure 3 shows that the experimental data for the cases of coated TIM are compared with the Intel data, the data distribution in Figure 3 reveals that, as the fan speed increased, the temperature $T_{C1}$ of the CPU simulator significantly decreased. When the fan speed was increased from 2200 rpm to 2400 rpm, the average reduction in $T_{C1}$ reached 2.98% and after adjustment to 2600 rpm, the rate increased to 5.8%. Under heat load $Q=87.6$ W, the fan speed was increased from 2200 rpm to 2600 rpm, and static loads of 12 kgf and 18 kgf, the effective temperature reductions were 7.3% and 4.8%, respectively.

Analysis of the data distribution shown in Figure 4 showed that the value of the thermal characterization parameter $\psi$ became smaller as the fan speed increased, which was consistent with the Intel data [25]. Intel reported that, in the reference design performance of Intel RCBFH-3, the thermal characterization parameter $\psi$ is 0.29 °C/W at 3600 rpm; and 0.325 °C/W at 2400 rpm, with the heat-reducing rate reaching 12.07%. In our experiment, due to the coating of TIM on the contact interface, the reduction of the contact resistance is comparatively more significant. Therefore, the value of the thermal characterization parameter $\psi$ showed a lower amount of contact-resistance reduction.

Finally, we explored the impact of the fan speed on the conductive heat flux, as shown in Figure 5. The experimental results clearly show that with an increase in the heat load $Q$, the conductive heat flux $q''$ showed a linear increasing trend. This implies that heat transfer through the copper core increased. Comparing the data obtained at the two different fan speeds (2200 and 2400 rpm) at the same static load of 18 kgf, it is obvious that the faster the fan speed, the more is the heat flux $q''$, especially under conditions of high heat load $Q$.

4. Conclusion

The results of this experimental study on the heat-dissipative performance of the heat sink for a desktop PC can be used to select thermal solutions that optimize heat-dissipative performance. This study is the first to analyze the interrelationships among static load, TIMs, and fan speed. The following conclusions are drawn from the analysis:

The results of the experiment with or without TIM coating clearly show that the temperature $T_{C1}$, which corresponded with heat dissipation, significantly decreased. At a fan speed of 2600 rpm and static load $F=18$ kgf, the largest reduction of $T_{C1}$ reached 43.8%, which indicates that applying thermal grease weakened the contact resistance and facilitated the transfer of heat from the interface to the heat sink. Thus, it significantly reduced the temperature of the CPU simulator and upgraded the electric power of the CPU to a higher range.

Generally, a greater static load reduces the temperature of the CPU simulator. The temperature reduction results from enlargement of the contact pressure between the CPU and the heat sink due to the load. Thus, it substantially increases the squeeze on the metal and results in the reduction of air gap between the contact surfaces.
In the context of static load, this study showed that the temperature of the CPU simulator changed significantly (less than 4.2% of the maximum) in spite of the increase in the static load from 12 kgf to 18 kgf (50% increase). Nevertheless, when the static load reaches a larger value, the temperature $T_{CI}$ of the CPU simulator is reduced.

This study explored how the thermal characterization parameter $\psi$ reflects the entire heat-dissipative performance. The experimental results show that a larger static load can enlarge the pressure on the interface between the CPU and the heat sink. The compression of the metal reduces both the air gap between the contact surfaces and the thermal contact resistance. Thus, it increases heat dissipation and reduces the thermal characterization parameter $\psi$. However, at higher fan speeds, increased heat dissipation caused by forced convection results in a relatively smaller thermal characterization parameter $\psi$. Compared to static load, the fan speed has a relatively larger effect on the thermal characterization parameter $\psi$.

Finally, the experimental results indicated that, as the heat load $Q$ increased, the conductive heat flux $q''$ increased, which implies that heat conduction through the copper core has increased. While assessing the impact of the static load and the fan speed, we found that the greater the static load, the larger is the heat flux $q''$ conducted through the copper core; the faster the fan speed is, the larger is the heat flux $q''$. The impact of fan speed on the heat-flux conduction $q''$ was significantly greater than the impact of static loads.

In summary, the most direct way to solve the CPU-heating problem is to simplify the attachment mechanism between the CPU and the heat sink to ensure the effectiveness of the TIM between the CPU and the heat sink. This solution is simpler and more economical than improving the heat sink fins.

Reference


