Experimental investigations of thermal resistance of a heat sink with horizontal embedded heat pipes[☆]

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Abstract

This article experimentally investigates the thermal resistance of a heat sink with horizontal embedded heat pipes. Heat sink with embedded heat pipes disperses heat from CPU to both the base plate and the heat pipes, and then removes heat from fins to the surrounding. This experimental approach measures the thermal resistances of interface between CPU and base plate, base plate, CPU to heat pipes, heat pipes and fins through the thermal resistance analysis. It can be divided into two steps. The first step is to measure the thermal performance of a heat pipe and the next step is the measurement for the thermal performance of heat sink with and without the function of heat pipes. These results are limited to the case of sink-processor assemblies installed horizontally and two U-shaped embedded heat pipes inserted into the heat sink in the paper. The results show that two heat pipes embedded in the base plate carry 36% of the total dissipated heat from CPU, while 64% of heat is delivered from the base plate to the fins. Furthermore, when the CPU power is 140 W, the total thermal resistance is at its minimum 0.27 °C/W and the thermal resistances of CPU to heat pipes and heat pipes are 0.32 °C/W and 0.12 °C/W respectively.

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1. Introduction

The performance in electronic components has been increasing, resulting in the problem of increased operating wattage and power [1]. The normal operating temperature of a silicon chip is below 70 °C. Bar-Cohen et al [2] pointed out that the reliability of electronic components decreases by 10% for increasing every 2 °C above normal operating temperature. If the operating temperature of the chip is too high, the stability and performance of electronic components decreases. This is the principal reason for the failure and reduced durability of electronic components. Thus, it is necessary to quickly remove heat generated by electronic components for normal operation.

Typical thermal solution of electronic components heat dissipation is to install a heat sink with a fan, as shown in Fig. 1a. Heat is transferred from the base plate to the fins and removed from fin surfaces to environment through forced

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Nomenclature

A	area, m ²
A_e	contact surface area of the solder paste, m ²
h	heat transfer coefficient, W/m ² K
H	height from base plate to condensation section of heat pipes, m
H_f	fin height, m
H_1	height from base plate to adiabatic line, m
H_2	height from adiabatic line to condensation section of heat pipes, m
k	thermal conductivity, W/mK
k_f	fin thermal conductivity, W/mK
n	number of heat pipes
Q	total heat transfer rate, W
Q_1	heat transfer rate from base plate to fins, W
Q_2	heat transfer rate from heat pipes to fins, W
Q_3	heating power of single heat pipe, W
R	thermal resistance, K/W
R_t	total thermal resistance, K/W
R_c	thermal contact resistance, K/W
R_n	fin-base convective thermal resistance, K/W
R_h	base to heat pipes thermal resistance, K/W
R_p	heat pipes thermal resistance, K/W
R_f	fin-pipe convective thermal resistance, K/W
R_s	spreading thermal resistance, K/W
R_m	conduction thermal resistance, K/W
R_3	single heat pipe thermal resistance, K/W
t	thickness, m
t_f	fin thickness, m
t_h	thickness from CPU to heat pipes, m
T	temperature, K
T_u	mean upper surface temperature of base plate, K
T_b	one-dimensional temperature of adiabatic line, K

Greek symbols

α	modified base plate
β	modified base to heat pipes

Subscripts

a	ambient
b	base plate
c	condensation section of heat pipes
d	lower surface of base plate
e	evaporation section of heat pipes
g	thermal grease
h	heat source
s	solder paste

convection. Currently, increasing the fin surface area and raising fan speed are keys to solve the high heat power problems. The total thermal resistance is used to evaluate the thermal performance of a heat sink. Duan and Muzychka [3] investigated four different sizes of heat sinks and show that increasing the fin surface area reduces the total thermal

Fig. 1. (a) Heat sink without heat pipes. (b) Heat sink with embedded heat pipes.

resistance from 0.55 °C/W to 0.35 °C/W. Lin et al. [4,5] studied the effect of fan speed on thermal resistance. The results show that a heat sink with a maximum fan speed of 4000 RPM has an optimum total thermal resistance value of 0.33 °C/W. However, increasing the surface area results in an increase in cost and boosting the fan speed results in noise, vibration and more power consumptions, which increase the probability of failure to electronic components.

Fig. 1b shows a longitudinally-finned heat sink with embedded heat pipes which uses two heat pipes, one end of the heat pipes inserted into the base plate which wrap around 180° as the evaporation section and the other end protrude and incline about 30° angle through the upper portion of the fins as the condensation section. It disperses heat from heat source to both the base plate and the heat pipes embedded in the base plate, and then removes heat from fins to the surrounding. The heat pipe is composed of a container making up highly pure oxygen-free copper pipe; it includes a wick structure and working fluid [6–8]. The working fluid evaporates when heating, and then condenses through cooling. The wick structure can offer capillary force for working fluid flow to oppose gravity [9]. And the maximum heat dissipation is affected by the inclinations of embedded heat pipes reduced about 20% between +90 to 0° opposition gravity [10,11]. Due to high thermal conductivity and so-named gravity-assistance of the heat pipes, the heat sink with embedded heat pipes can reduce the total thermal resistance to under 0.3 °C/W [12,13]. Xie et al. [14] conducted an experiment combining a 4-mm diameter heat pipe and a heat sink, achieving an optimum total thermal resistance of 0.29 °C/W. Gernert et al. [15] employed an experiment on a heat sink with embedded heat pipes with a 25.4-mm diameter, 156 mm long heat pipe. At a maximum heat flux of 285 W/cm², it has the lowest total thermal resistance 0.23 °C/W. Therefore, the heat sink with embedded heat pipes is one of the best solutions for thermal problems in electronic components.

Previous research only measures the interface, base plate and total thermal resistances of a heat sink with embedded heat pipes and fails to provide the proportion of the heat borne by the heat pipes. In light of this, present research examined the thermal performance of the heat sink with and without the function of heat pipes and calculated the amount of heat transferred from the base plate to the fins and from the embedded heat pipes to the fins respectively. The individual thermal resistances of contact, base plate, base to heat pipes, heat pipes, fins and the total thermal resistance can be obtained through thermal resistance analysis. The ratio of heat transferred through the heat pipes to the fins can be also acquired.

2. Thermal resistance analysis

This study consider heat sink with embedded heat pipes as steady one-dimensional heat flow which CPU surface temperature and air environment temperature maintained constant as T_h and T_a respectively. The rate of heat transfer Q from the CPU (heat source) to the surrounding air can be expressed as

$$Q = \frac{\Delta T}{R_t} \quad (1)$$

Fig. 2. (a) Heat sink with embedded heat pipes heat flow pathways. (b) Fin temperature distribution.

where R_t denotes the total thermal resistance of heat sink with embedded heat pipes. Fig. 2a shows that heat transfer rate Q transfers heat from the heat source to the fins through the base plate and the heat pipes. The base plate transfers heat to fins is Q_1 and the heat pipes transfer heat to fins is Q_2 . Then

$$Q = Q_1 + Q_2 \quad (2)$$

The temperature profile along the fins is shown in Fig. 2b. The region where temperature gradient equals to zero appearing between these two heat transfer pathways is called adiabatic line. The fins in the adiabatic line are similar to be insulated. It means that no heat flow across the adiabatic line. If the temperatures are assumed uniform both on the upper surface of base plate and on the condensation section of heat pipes, the steady one-dimensional energy balance equation for fins is [16]

$$\frac{d^2 T}{dY^2} - m^2(T - T_a) = 0 \quad (3)$$

and

$$m = \sqrt{\frac{2h}{k_f t_f}}, \quad (4)$$

where h represents the convective heat transfer coefficient, k_f is the fin conductivity and t_f is the fin thickness. The specified boundary conditions can be expressed as

$$\begin{aligned} T &= T_u \text{ at } Y = 0 \text{ and} \\ T &= T_c \text{ at } Y = H, \end{aligned} \quad (5)$$

In Eq. (5) T_c is the temperature of the condensation section of heat pipes, T_u is the average temperature of the upper surface of base plate and H denotes the length between base plate and condensation section of heat pipes. The temperature profile for the fins can be obtained from Eq. (3) as

$$T(Y) = T_a + \left[\frac{(T_u - T_a)e^{mH}}{e^{mH} - e^{-mH}} - \frac{T_c - T_a}{e^{mH} - e^{-mH}} \right] e^{-mY} + \left[\frac{(T_c - T_a)}{e^{mH} - e^{-mH}} - \frac{(T_u - T_a)e^{-mH}}{e^{mH} - e^{-mH}} \right] e^{mY} \quad (6)$$

From the definition of adiabatic line,

$$\frac{\partial T}{\partial Y} = 0 \text{ at } Y = H_1, \quad (7)$$

Fig. 3. Thermal resistance analysis network.

The location of adiabatic line can be shown as

$$H_1 = \frac{1}{2m} \ln \left[\frac{(T_u - T_a)e^{mH} - (T_c - T_a)}{(T_c - T_a) - (T_u - T_a)e^{-mH}} \right] \quad (8)$$

Since there is no heat across the adiabatic line, the thermal resistance network for heat transfer through the heat sink with embedded heat pipes can be shown in Fig. 3. The Q_1 path includes the base plate resistance R_b and fin-base convection resistance R_n ; the Q_2 path includes the base to heat pipes resistance R_h , heat pipes resistance R_p and fin-pipe convection resistance R_f . The total thermal resistance can be expressed as the sum of the thermal contact resistance and those thermal resistances on the heat flow paths of Q_1 and Q_2 , which is

$$R_t = R_c + \frac{1}{\frac{1}{(R_b + R_n)} + \frac{1}{(R_h + R_p + R_f)}} \quad (9)$$

In Eq. (9), contact resistance R_c is defined as the effective temperature difference at the interface (the temperature of heat source T_h minus the temperature at the center of the lower surface of the base plate T_d) divided by total Q . The thermal contact resistance can be minimized by applying thermal grease, such as silicon oil, on the contact surface before they are pressed against each other. The thickness of the thermal grease is t_g , the thermal conductivity is k_g and the contact surface area is A_g , which are related to the thermal contact resistance by

$$R_c = \frac{t_g}{k_g A_g} \quad (10)$$

Base plate resistance is defined as the temperature difference between the center of the lower surface of base plate (T_d) and the average temperature of the upper surface of base plate (T_u) divided by Q_1 . This thermal resistance is produced by the thickness of base plate and the thermal properties of the material. If the base plate is assumed as a plate with no heat pipes embedded, the base plate resistance including the spreading resistance R_s , the conduction resistance R_m and the modified base resistance R_α can be expressed as

$$R_b = R_m + R_s + 2 \cdot R_\alpha \quad (11)$$

Conduction resistance can be determined from equation below

$$R_m = \frac{t_b}{k_b A_b} \quad (12)$$

where t_b is the thickness of the base plate, k_b is the thermal conductivity of base plate and A_b is the plate area. The spreading resistance from Lee et al. [17] can be used as

$$R_s = \frac{\sqrt{A_b} - \sqrt{A_h}}{k_b \times \sqrt{\pi A_b A_h}} \times \frac{\lambda k_b A_b R_n + \tanh(\lambda t_b)}{1 + \lambda k_b A_b R_n \tanh(\lambda t_b)} \quad (13)$$

$$\lambda = \frac{\pi^{1.5}}{\sqrt{A_b}} + \frac{1}{\sqrt{A_h}} \quad (14)$$

where A_h is the heat source area.

The $2.R\alpha$ is a fixed value that equals to the thermal resistance difference between the base plate resistance and the sum of the conduction resistance and spreading resistance. And it will be zero when the base plate had not been inserted in the heat pipes.

Fin-base convective resistance is defined as the average temperature of the upper surface of base plate (T_u) minus the ambient temperature (T_a), divided by the heat transfer rate Q_1 , which can be determined from

$$R_n = \frac{1}{h\eta_1 A_1} \quad (15)$$

$$\eta_1 = \frac{\tanh(mH_1)}{mH_1} \quad (16)$$

In Eq. (16) h is the convective heat transfer coefficient, η_1 is the surface efficiency at fin length H_1 and A_1 is the fin area up to the height of H_1 .

Base to heat pipes resistance is defined as the temperature difference between the center of the lower surface of base plate (T_d) and the evaporation section (T_e) of heat pipes divided by the heat transfer rate Q_2 . This thermal resistance includes the conduction resistance of the medium from the heat source to the heat pipes and the contact interface between the base plate and the heat pipes. Solder paste is applied to affix the heat pipes within the base plate at the surface contact. It is the material of commercial solder silver with 42 °W/mK. Thus the base to heat pipes resistance can be expressed as

$$R_h = \frac{t_h}{k_b A_h} + \frac{t_s}{k_s A_e} + R_\beta \quad (17)$$

where t_s is the thickness of the solder paste, k_s is the solder paste's thermal conductivity, A_e is the contact surface area of the solder paste, t_h is the distance from CPU to heat pipes and R_β is the modified base to heat pipes resistance.

The variable value of R_β is the thermal resistance difference between the base to heat pipes resistance and the sum of the conduction resistance of the medium from the heat source to the heat pipes and the contact interface resistance between the base plate and the heat pipes. It will be change with the performance of embedded heat pipes and the embedded distance between the two heat pipes.

Heat pipes resistance is defined as the temperature differences between the evaporation section (T_e) and the condensation section (T_c) divided by Q_2 , which can be determined from the thermal performance of single heat pipe experiments [18]. Fin-pipe convective resistance is defined as the temperature differences between the condensation section of heat pipes (T_c) and the ambient temperature (T_a) divided by Q_2 . Similar to fin-pipe convective resistance, it can be expressed as

$$R_f = \frac{1}{h\eta_2 A_2} \quad (18)$$

$$\eta_2 = \frac{\tanh(mH_2)}{h\eta_2 A_2} \quad (19)$$

In Eq. (19) h is the convective heat transfer coefficient along the region of fin length H_2 , η_2 is the corresponding surface efficiency and A_2 is the fin surface area for the height of $(H_f - H_1 - d_{hp})$, where d_{hp} is the diameter of the heat pipe.

3. Experimental investigation

Experimental investigation is divided into two parts: the first is the thermal performance experiment for single heat pipe; the second is the thermal performance experiments for the heat sink with and without the function of heat pipes. Fig. 4 shows the experimental apparatus for a heat pipe. The heat pipe is a bending pipe with a diameter 6 mm and a total length of 170 mm. The length of evaporation section and condensation section is 65 mm respectively, and the insulated length is 40 mm. The materials of heat pipe's container and wick type are copper metal and sintered structure of pure copper powder respectively. The evaporation section employs a dummy heater connecting with the power supply unit in order to simulate CPU power. The water from the cooling circulator flows through the water jacket to control the temperature at the condensation section. Thermocouples are attached to the evaporation and condensation sections to measure their average temperatures. Heat pipe, dummy heater and water jacket are encased with a foaming agent to insulate the system. The experimental parameters are conducted at heating power levels of 1 W, 5 W, 7 W and so on up to 55 W, until the evaporation section is dried out.

Fig. 5 shows the experimental apparatus for the heat sink with embedded heat pipes. The upper surface of the dummy heater is coated with thermal grease to reduce contact resistance. A thermocouple is attached to the upper surface of the dummy heater to measure the temperature (T_h). Six thermocouples are attached to the center of the lower surface of base plate and five points along the diagonal of the upper surface of base plate, measuring the temperatures at the center of the lower surface of the base plate (T_d) and the average temperature of the upper surface of the base plate (T_u). Computational Fluid Dynamic (CFD) software (Icepak software) is used to calculate the average temperature of T_u , which compares with the experimental measurement of five-point average value and develops a correlation between them within an error of $\pm 3\%$. A fan with 4200 RPM and 46 CFM is placed to the top of heat sink to disperse heat through forced convection. A thermocouple is placed to the top of the fan to measure the ambient temperature (T_a). Thermocouples are attached to the evaporation and condensation sections of heat pipes to measure the temperatures of T_e and T_c . All thermocouples are T-type. This experiment starts with a heating power of 60 W and stops 200 W increasing it by increments of 20 W.

After completing the above experiment with function of heat pipes, let these two heat pipes in heat sink fail to function. The method for letting these two heat pipes fail to function is to split the middle position between evaporation and condensation section of two embedded heat pipes. After splitting heat pipes, the heat sink with embedded heat pipes would be heated over 100°C , and the working fluid escaped from these pipes. The heat sink experiments without the function of heat pipes are performed as same as that with the function of heat pipes. Then the corresponding thermal resistances can be determined from this experimental analysis.

The thermocouples used in the experiment have a measurement error of $\pm 0.5^\circ\text{C}$. The cooling circulator manufactured by Firstek Scientific Ltd., Co. has a measurement error of $\pm 0.5^\circ\text{C}$. The data recorder manufactured by

Fig. 4. Experimental apparatus for single heat pipe.

Fig. 5. Experimental apparatus for Heat sink with embedded heat pipes.

4. Results and discussion

Fig. 6 shows the thermal resistance of a heat pipe with different power input. The heat pipe is unable to operate effectively at low heating power of 5 watts or less, resulting in a high thermal resistance. The thermal resistance has dropped from 0.49 °C/W to 0.33 °C/W at 7 W, signifying that the working fluid within the heat pipe has started to operate. From a heating power of 15 to 30 W, the working fluid within the heat pipe is operating efficiently, lowering the thermal resistance to below 0.3 °C/W. The total thermal resistance reaches a minimum of 0.24 °C/W when the heating power is 25 W. The heat pipe is considered to have a possible local dry-out limit for heat inputs higher than 30 to 35 W. At high heating power of 35 to 50 W, it results in higher vapor temperature and

Fig. 6. Performance curve for heat pipe total thermal resistance under various heating power.

Fig. 7. Illustration for bypass heating power.

pressure, and the noncondensable gas is compressed more toward the end of the condensation section. Thus the thermal resistance increases from 0.35 °C/W to 0.94 °C/W. The temperature at the evaporation section surpasses 100 °C when the heating power is over 55 W. At this time the heat pipe has burned out and lost its effectiveness. From above-mentioned, the thermal load higher than 40 W does not consider long-term or tilted application in this study. From the experimental results we can derive the correlation between heating power and heat pipe resistance as

$$R_3 = 0.324 + 0.013(Q_3) - 0.002(Q_3)^2 + 8.079 \times 10^{-5}(Q_3)^3 - 8.776 \times 10^{-7}(Q_3)^4 \quad (20)$$

Fig. 7 shows the experimental results of thermal performance lines for a heat sink with and without the function of heat pipes. As the heating power is increasing, the temperature difference ($T_u - T_a$) between the upper surface of the base plate and the surrounding air is also increasing no matter with or without the function of heat pipes. The slopes of these two lines represent the thermal resistances. Because the slope of the upper line is greater than that of the lower, the thermal resistance without function of heat pipes is greater than that with function of heat pipes. It means that the temperature difference ($T_u - T_a$) is higher without the function of heat pipes than that with the function of heat pipes. This is because of having an extra path transfer's heat from heat pipes to surrounding and thus the temperature difference is smaller. As the temperature difference for the two lines is fixed as shown in Fig. 7, it reaches the thermal performance lines with two points at the power input with and without function of heat pipes. The corresponding heating power represents the Q_1 in the heat sink without the function of heat pipes and the total Q with the function of two embedded heat pipes. Heat transfer rate Q_2 is equal to the total Q minus the Q_1 . As indicated in Fig. 7, the temperature difference is 13.3 °C, the heating power for Q is 140 W, and the power for Q_1 is 88.2 W. Therefore Q_2 equals 51.8 W. Table 1 shows the ratio of bypass

Table 1
Ratio of bypass heating power and to total heating power

Q (W)	Q_1/Q (%)	Q_1 (W)	Q_2/Q (%)	Q_2 (W)
60	67	40.2	33	19.8
80	64	51.2	36	28.8
100	65	65	35	35
120	63	75.6	37	44.4
140	63	88.2	37	51.8
160	64	102.4	36	57.6
180	64	115.2	36	64.8
200	65	130	35	70

Fig. 8. Relationship of the total thermal resistance and contact resistance with the heating power Q .

heating power to total heating power. The average ratio Q_1/Q is 64% and the average ratio Q_2/Q is 36% for such a heat sink with two embedded heat pipes between 60 and 200 W.

Fig. 8 shows the total thermal resistance and thermal contact resistance with heating power Q . The experimental total thermal resistance falls from 0.30 °C/W to 0.27 °C/W when the heating power is 60 to 140 W and reaches its minimum of 0.27 °C/W at 140 W. The reason is when the heating power increases to the point that the two embedded heat pipes are starting to function, the total thermal resistance shows a decreasing trend. The two embedded heat pipes are unable to withstand higher heat at heating powers of above 200 W, causing the higher vapor temperature and pressure, and the noncondensable gas is compressed more toward the end of the condensation section, resulting in the heat pipes losing performance, thereby increasing total thermal resistance. The theoretical total thermal resistance has the same curve trend with the experimental total thermal resistance. The value of total thermal resistance changes with power [19] because the heat sink with two embedded heat pipes includes two-phase heat dissipation device as heat pipe. Fig. 8 also shows the theoretical results from thermal resistance analysis and experimental measurements. The experimental contact resistance is approximately 0.03 °C/W when the heating power is between 60 and 200 W and is close to the theoretical predictions; therefore the contact resistance in this experiment can be considered a constant.

The ratios of bypass heating power Q_1/Q and Q_2/Q are changed with the total thermal resistance. As shown in Table 2, when the value of the theoretical Q_1 -path thermal resistances (e.g. $R_b + R_n$) is 0.4 °C/W, the ratio of bypass heating power Q_1/Q changes with the total thermal resistance. But the ratio of bypass heating power Q_2/Q changes not only with the total thermal resistance but the theoretical Q_2 -path thermal resistances (e.g. $R_h + R_p + R_f$). The Q_2 -path thermal resistances affect the total thermal resistance because of two embedded heat pipes inserted into the heat sink. If there is a very good thermal performance of heat pipe using in this

Table 2
Thermal resistances versus the ratio of bypass heating power

R_t (°C/W)	Q_1/Q (%)	$R_b + R_n$ (°C/W)	Q_2/Q (%)	$R_h + R_p + R_f$ (°C/W)	R_b (°C/W)
0.3	67	0.4	33	0.77	0.12
0.29	64	0.4	36	0.7	0.079
0.28	65	0.4	35	0.69	0.075
0.27	63	0.4	37	0.65	0.045
0.27	63	0.4	37	0.64	0.036
0.27	64	0.4	36	0.67	0.041
0.28	64	0.4	36	0.71	0.056
0.29	65	0.4	35	0.7	0.068

Fig. 9. Relationship of the base plate resistance and the fin-base convective resistance with the heating power Q_1 .

experiment, the ratio of bypass heating power Q_2/Q will be over 36% under fixed value of Q_1 -path thermal resistances. The maximum Q_2/Q is 46% while R_p and R_f are zero respectively in the paper. This means that using a perfect heat pipe with zero thermal resistance embedded in base plate of heat sink will reach their maximum Q_2/Q and minimum Q_1/Q in this study.

Fig. 9 indicates the base plate resistance and fin-base convective resistance with heating power Q_1 . The experimental base plate resistance and fin-base convective resistance are approximately 0.25 °C/W and 0.15 °C/W respectively when Q_1 is between 40 and 129.5 W, not changing as heating power increases. Thus the base plate resistance and fin-base convective resistance in this experiment can be considered constants. The sum of R_m and R_s is 0.16 °C/W from Eqs. (12) and (13), thus the $R\alpha$ equals 0.045 °C/W. And the theoretical and experimental base plate resistances become closer as shown in Fig. 9. The reason is that the components transferring heat through this path of heating power Q_1 transferred from the base plate without function of heat pipes to the fins are

Fig. 10. Relationship of the base to heat pipes resistance and the fin-pipe convective resistance with the heating power Q_2 .

Fig. 11. Relationship between the heat pipes resistance and the heating power Q_2 .

all solid. The thermal physical properties of these components are the same when there is not much change in temperature. Thus the base plate resistance and fin-base convective resistance should remain constant. The base plate resistance and base-fin convective resistance obtained in this experiment are both constant, proving that the experimental results are correct.

Fig. 10 indicates the base to heat pipes resistance and fin-pipe convective resistance with heating power Q_2 . The experimental base to heat pipes resistance drops from 0.41 °C/W to 0.33 °C/W under heating power of 20 to 44 W. When Q_2 is 51.4 W, it reaches a minimum value of 0.32 °C/W. Because the theoretical base to heat pipes resistance takes into account the value of R_b as shown in Table 2 and the thermal performance of the embedded base plate heat pipe, the theoretical base to heat pipes resistance is consistent with the experimental result. The experimental fin-pipe convective resistance is approximately 0.20 °C/W when the Q_2 is 20 to 70.5 W and does not change much as the heating power increases. Moreover, the theoretical and the experimental fin-pipe convective resistances are close, so the convective resistance R_f can be considered a constant in this experiment.

Fig. 11 indicates the relationship between the heat pipes resistance and heating power Q_2 . The experimental heat pipes resistance drops from 0.16 °C/W to 0.13 °C/W when Q_2 is between 20 and 44 W. It reaches its minimum value of 0.12 °C/W when Q_2 is 51.4 W. The experimental heat pipes resistance rises from 0.15 °C/W to 0.19 °C/W when Q_2 is between 58.3 and 70.5 W, increasing as the heating power increases. The dummy heater was placed at the center of the base plate and these two embedded heat pipes in the heat sink were laid out symmetrically on the left and right, so it can be assumed that these two heat pipes embedded in base plate each loaded an equal amount of heat. In other words, the Q_2 is twice the Q_3 using in the Eq. (20). When Q_2 is 51.4 W, Q_3 is 25.7 W. The experimental results of single heat pipe show that this value is within the heating power range of 20 to 27 W which it is at minimum thermal resistance value, proving the validity of this experiment. The theoretical and experimental heat pipes resistances coincide with each other closely, meaning that the heat pipes resistance of heat sink with embedded heat pipes can be obtained from the thermal resistance of the parallel single heat pipe.

5. Conclusions

The present analysis is only valid for the sink-processor assemblies installed horizontally. The two embedded heat pipes used in the paper consisted of pure copper container and sintered copper powder with water as the working fluid. Through the experimental method and thermal resistance analysis of this study, it can be obtained that heat transfer rate Q_1 and Q_2 account for 64% and 36% of the total Q respectively. But if it is considered in different orientations with respect to gravity or different performance (e.g. wick structure, working fluid), the results may be different. Moreover, the total thermal resistance of the heat sink with embedded heat pipes is only affected by changes in the base to heat pipes resistance and heat pipes resistance over the heat flow path of the Q_2 ; that is, the total thermal resistance changes with the quality of the performance of heat pipes.

The theoretical total thermal resistance is in good agreement with the experimental total thermal resistance. The curve of theoretical heat pipes resistance is consistent with curve of experimental heat pipes resistance, verifying that the experimental method and thermal resistance analysis used in this article are valid. Thus the measuring and analytical methods for the thermal performance of the heat sink with embedded heat pipes are completely established in the present study. The maximum Q_2/Q and minimum Q_1/Q are 46% and 54% respectively for such a heat sink in this paper. The model can be used to help the design of the heat sink similar to the one investigated herein.

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