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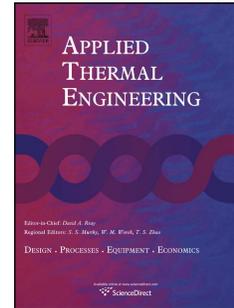
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Characterization of cooling systems based on heat pipe principle to control operation temperature of high-tech electronic components

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Abstract

The use of cooling systems based on heat pipe principle to control operation temperature of electronic components is very efficient. They have an excellent miniaturizing capacity and this fact creates adaptability for more practical situations. Starting from the observation that these cooling systems are not precisely characterized from the thermal efficiency point of view, the present paper proposes a methodology of data acquisition for their thermal characterization. An experimental set-up and a data processing algorithm are shown to describe the cooling of a heat generating electronic device using heat pipes. A Thermalright SI-97 PC cooling system is employed as a case-study to determine the heat transfer characteristics of a fins cooler.

Keywords: Heat pipe; Heat transfer; Electronic components cooler; Fins cooler

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1. Introduction

Cooling systems for personal computer (PC) electronic components consist of simple devices, e.g. metal heat sinks, or of advanced systems, such as liquid-cooled heat plates, heat pipes, microchannel heat sinks etc. Generally speaking of a PC (a desktop or a notebook), it is important to maintain the operation temperature of the central processing unit (CPU) below a certain level, otherwise the PC reliability is compromised [1]. Miniature heat pipe coolers fitted with a heat sink are an optimum choice, taking into account both cost and performance, due to their simple structure, lack of active elements needed for working fluid transport and high efficiency of heat transfer [2-11].

A heat pipe transfers heat by boiling and condensation of a working fluid in a fully-enclosed vacuum pipe, having an effective conductivity several times or even nearly 10 000 times better than that of good heat-conducting materials (copper, silver etc) [12]. Figure 1 illustrates the principles of a gravitational and a wick heat pipe, respectively. The vapour produced in the boiling section (evaporator) flows rapidly into the condensation section (condenser), wherein it is cooled and turned into a liquid, which flows back into the boiling section (by free fall or through the wick structure).

Besides PC electronic components cooling, heat pipes are also employed in space technology [5,13], equipments for waste heat recovery (air preheaters, steam generators) [10,12], cooling units [10,14,15], implanted neural stimulators, sensors, pumps and temperature warning systems [11].

The challenge of CPU cooling consists of restricting its temperature to 60-90 °C at a heat sink temperature of 35-45 °C [9]. A heat pipe itself cannot provide the cooling. It is, as a rule, only part of an integrated system, which ensures heat collection, transport

and dissipation. Accordingly, the hot end of the heat pipe is usually embedded into a heat spreader which is applied on the CPU surface by means of a silicone adhesive and a heat sink is assembled on the cold end.

An important parameter characterizing the efficiency of this complex cooling system is the total thermal resistance. It is defined as the temperature difference between the evaporator and the heat sink divided by the dissipated heat flow rate, Q . The total thermal resistance is a function of a large number of variables, such as heat pipe geometry and material, wick structure, working fluid properties, evaporator and condenser length [2,4,8-11]. There are several types of wick structures, e.g. metal (nickel, copper, titanium) sintered powder, groove, fine fibre bundle and screen mesh. Two main properties of the wick are the pore size (radius) and the permeability. The pore size determines the fluid pumping pressure (capillary head) and the permeability is responsible for frictional losses of the working fluid. It was proved that the metal sintered powder wick, having a high fluid pumping pressure and a low permeability, is the most effective from a heat transfer point of view [11]. Water, ammonia, acetone, propane and toluene are usually used as working fluids. The temperature difference between evaporator and heat sink increases with heat flow rate, Q , whereas the total thermal resistance decreases with Q [2,8-10]. Thermal performance of a heat pipe, expressed by high values of Q , is improved at large values of diameter and inclination angle as well as at low values of length and wall thickness [8,11].

The problem of heat transport obstruction owing to the small space inside a notebook PC can be solved by miniature heat pipes fitted with fins (Figure 2). The design of a complex electronic device (desktop, notebook etc) imposes the control of its internal temperature field (very important for device performances). It can be correctly

assigned only if the operation characteristics of the used cooling system are known. The paper proposes an experimental set-up and a data processing algorithm which can be employed to establish the heat transfer characteristics of a microelectronic device cooler based on heat pipes fitted with a fins system.

2. Experimental

The main aims of the experimental work were to determine the characteristic operation curves of a heat pipe cooler in steady and unsteady state conditions and to establish the fins cooling system efficiency depending on the experimental measurements in steady state.

A Thermalright SI-97 model (Figure 3) was used as a CPU cooler in the experiments performed in the present study. This cooler was based on 4 heat pipes made of nickel-plated copper and containing acetone as a working fluid. The heat dissipation system (heat sink) consisting of $N=50$ aluminium fins (length $L=0.096$ m and width $l=0.028$ m) was assembled on the cold ends of heat pipes. The hot ends of heat pipes were embedded into a copper metallic piece (heat spreader), realizing an intimate contact between the pipes and the electronic component which was being cooled.

The experimental set-up in Figure 4 was employed to determine the functional characteristics of a cooling system. It permitted to measure accurately the heat flow rate and its dissipation in the adjacent medium by the fins system attached to the heat pipe cooler.

The set-up consists mainly of a heat supply system and a temperature values acquisition system. The heat supply system is composed of an aluminium bar (6) having cross-section dimensions of 30x30 mm and fitted with an electric resistance (7), which

is fed by an alternating current variator (10). The bar is thermally isolated by a glass cylinder (8) set into the supports (9). The Thermalright SI-97 cooler, consisting of heat pipes (1), a copper piece (2) and the fins system (3), is assembled at the upper part of the aluminium bar. Five thermocouples (4) are inserted into the aluminium bar, the first (T5) being situated at the interface between the bar and the copper piece and the others (T1-T4) at equal distances of 15 mm. Six thermocouples (5) – with T6 assembled on the cold end of a heat pipe and T7-T11 put on fins – measure temperature values in various zones of the heat dissipation system. All temperature values are continuously registered by a data acquisition system (14) and a computer (15). For measurements of forced convection, a cooling fan was assembled above the fins system. In this case, the heat flow rate transported by heat pipes was absorbed by an air stream up-flowing through the fins.

One experiment of natural convection and five experiments of forced convection at various values of fan rotor speed, n (500 rpm, 900 rpm, 1250 rpm, 2000 rpm and 2800 rpm), were performed for three values of heat flow rate, Q (34.8 W, 44.8 W and 54.8 W). Accordingly, the experimental plan contains 18 data sets resulting from six levels of air velocity and three levels of heat flow rate. Two sets of four coupled experiments were conducted at $Q=44.8$ W to emphasize the cooling system response when a switching between natural and forced convection ($n=2000$ rpm) occurred at its condenser.

3. Theoretical background

The heat flow rate transmitted through the aluminium bar at $z=0$ (interface between bar and hot ends of heat pipes) is equal to the heat flow rate transferred through

the fins system. It is determined according to the heat flux through the bar cross-section. The heat flux, q , supplied by the electric resistance of the experimental device to the cooling system, was obtained from a simplified Fourier equation [16,17]:

$$q = \lambda_{Al} \left(\frac{dt}{dz} \right)_{z=0} \quad (1)$$

where λ_{Al} is the aluminium bar thermal conductivity and $\left(\frac{dt}{dz} \right)_{z=0}$ the temperature gradient at the heat pipes surface.

The fins efficiency in steady state conditions, E , was estimated depending on the heat flow rate, Q , as follows:

$$Q = E\alpha S_t \Delta t = qS_b \quad (2)$$

where S_b is the bar cross-section area, α the heat transfer partial coefficient from fins to air, S_t the total transfer surface area of fins and Δt the difference between the fins and environment temperature. The fins surface area and the temperature difference were expressed by equations (3) and (4), respectively:

$$S_t = 2NLI \quad (3)$$

$$\Delta t = t_m - t_0 \quad (4)$$

where the mean temperature of the fins, t_m , is an average of the temperature values indicated by the thermocouples assembled on the fins ($t_m=(t_7+t_8+t_9+t_{10}+t_{11})/5$) and the measured environment temperature has a constant value ($t_0=20.7$ °C).

The fins efficiency was determined for set-up operation in natural and forced convection conditions in the fins system. The heat transfer partial coefficient, α , required by equation (2), was computed with the correlations corresponding to the flow in the fins space.

In the case of *forced convection*, the air volumetric flow rate, G_{va} , was obtained depending on the heat flow rate, Q , and the difference between air output and input temperature, $\Delta t_a = t_{a,out} - t_{a,in}$, from a heat balance equation:

$$G_{va} = \frac{Q}{\rho_a c_a \Delta t_a} \quad (5)$$

where ρ_a is the air density and c_a the air specific heat.

The air velocity, w , depends on the air volumetric flow rate, the fin length, the fins number and the distance between two fins, $d=0.0015$ m, as follows:

$$w = \frac{G_{va}}{Ld(N-1)} \quad (6)$$

Equations (7) and (8) were used to calculate Reynolds number, Re , and Prandtl number, Pr :

$$Re = wd_e \frac{\rho_a}{\eta_a} \quad (7)$$

$$Pr = \frac{c_a \eta_a}{\lambda_a} \quad (8)$$

where λ_a is the air conductivity, η_a the air viscosity and d_e the equivalent diameter of the air flow cross-section. The value of d_e was computed for the rectangular flow area formed between two adjacent fins with the following equation:

$$d_e = \frac{4Ld}{2(L+d)} \quad (9)$$

The Nusselt number, Nu , and the heat transfer partial coefficient of air, α , were determined according to equations (10) and (11), respectively [18-20]:

$$Nu = 0.664 Re^{0.5} Pr^{0.33} \quad (10)$$

$$\alpha = Nu \frac{\lambda_a}{d_e} \quad (11)$$

In the case of *natural convection*, the heat transfer partial coefficient of air, α , depends only on Δt and the characteristic length of solid surface, l , as shown in equation (12) [21]:

$$\alpha = 1.36 \left(\frac{\Delta t}{l} \right)^{0.25} \quad (12)$$

The following algorithm was used to obtain the characteristic parameters of heat pipe cooler in steady state conditions, for natural and forced convection:

Step 1. Input data: (a) the vectors giving the characteristic temperature dynamics of thermocouples T1-T11 were determined; (b) the z position vector of thermocouples T1-T5 was selected; (c) the speed of cooling fan rotor, n , was set; (d) the fins characteristics (N, L, l, d), the air parameters ($\rho_a, c_a, \lambda_a, \eta_a$), the aluminium bar parameters (S_b, λ_{Al}) and the temperatures $t_0, t_{a,in}$ and $t_{a,out}$ were specified;

Step 2. The temperature profile in the aluminium bar, $t(z) = ae^{bz}$ ($a, b = \text{const.}$), was plotted based on data set z_1-z_5 and $t_1(\tau)-t_5(\tau)$, in natural or forced convection conditions; the value of $\left(\frac{dt}{dz} \right)_{z=0}$ was determined and the input heat flux in the cooling system, q , was computed using equation (1);

Step 3. The time needed to achieve a steady state operation ($q = \text{const.}$), τ_s , was identified;

Step 4. The heat flow rate, Q , was determined with equation (2) depending on heat flux corresponding to steady state, q ;

Step 5. The driving force of heat transfer, Δt , was obtained using equation (4) based on mean temperature of fins, t_m , and environment temperature, t_0 ;

Step 6. The air velocity in the fins system, w , was computed with equations (5) and (6);

Step 7. Reynolds number, Re , and Prandtl number, Pr were obtained according to equations (7) and (8);

Step 8. Nusselt number was estimated in forced convection conditions by equation (10);

Step 9. The values of heat transfer partial coefficient of air, α , were determined with equation (11) for forced convection and equation (12) in natural convection conditions;

Step 10. The fins efficiency, E , was estimated by equation (2).

4. Results and discussion

As it was specified, 18 experiments of system heating dynamics investigation were conducted. The algorithm described at the end of chapter 3 was used for characteristic data processing of each experiment, in order to obtain the dependencies $E=E(w, Q)$ and $\tau_s=\tau_s(w, Q)$.

Figure 5 illustrates the experimental system heating dynamics when the cooler dissipates heat in natural and forced convection conditions ($n=2000$ rpm) in the surrounding air. According to steps 1-4 of the algorithm proposed, these curves correspond to a heat flow of 44.8 W. In natural convection conditions it is observed that the steady state temperature values are quite high. Accordingly, they are inadequate for a proper operation of an electronic device, such as a PC (e.g. temperature value at interface between aluminium bar and heat pipes, $t_5=78.2$ °C), which would result in its degraded performance. Moreover, the time needed to establish the steady state is double ($\tau_s=40$ minutes) than for forced convection ($\tau_s=20$ minutes). The value of interface temperature in forced convection conditions, $t_5=46.3$ °C, proves an enhancement of the

heat generating system cooling. Steady state data in natural and forced convection conditions for $Q=44.8$ W and various values of fan rotor speed are listed in Table 1.

The characteristic parameters of cooler in steady state conditions, namely τ_s , Δt_a , Δt , w , Re , Nu , α and E , were computed according to the algorithm steps described in the paragraph 3. The results obtained for six values of fan rotor speed, n (0 rpm, 500 rpm, 900 rpm, 1250 rpm, 2000 rpm and 2800 rpm), and three values of heat flow rate, Q (34.8 W, 44.8 W and 54.8 W), are shown in Table 2. Data summarized in Table 2 emphasize the following aspects:

- the time needed to achieve a steady state operation, τ_s , estimated at step 3, is shorter at higher values of n and Q ;
- the differences between air output and input temperature, Δt_a , and between fins and environment temperature, Δt , determined at steps 1 and 5, respectively, increase when n decreases and Q increases;
- the air velocity in the fins system, w , the numbers Re and Nu , the heat transfer partial coefficient of air, α , and the fins efficiency, E , established at steps 6-10, are enhanced at high values of n and Q ;
- at the same value of Q , the values of E and τ_s estimated for natural convection ($n=0$ rpm) are near the values obtained in forced convection conditions at the lowest value of fan rotor speed ($n=500$ rpm).

The curves depicted in Figure 6, obtained according to the data in Table 2, illustrates fins efficiency, E , and time needed to establish the steady state, τ_s , depending on air velocity, w , and heat flow rate, Q . As can be seen, efficiency increases and time decreases with air velocity and heat flow rate.

An important problem concerning the cooler operation is to maintain the temperature of interface between the heat generating system and the heat spreader, t_5 , in a certain range, e.g. 45-65 °C. For this purpose, the establishment of cooler fan command to switch between natural convection (NC) and forced convection (FC) was proposed.

Figure 7 shows the temperature dynamics obtained for four coupled cooling experiments in NC and FC ($n=2000$ rpm) conditions for $Q=44.8$ W. The system operation was conducted until the steady state was attained in each experiment, i.e.: 50 minutes in NC ($\tau_s=40$ min), followed by 20 minutes in FC ($\tau_s=14$ min), then 50 minutes in NC ($\tau_s=40$ min) and again 20 minutes in FC ($\tau_s=14$ min). The transition between the both convection modes was performed by starting or stopping the cooler fan. Detailed information on time variation of interface temperature, t_5 , in these experiments is listed in Table 3.

The first stage of operation in NC conditions begins with a thermal inertia period ($t_5 \approx 21.0$ °C) of 6 minutes which can be considered as a steady state. It is followed by an unsteady heating during 34 minutes and a steady state ($t_5 \approx 78$ °C) during the next 10 minutes. At minute 50 the fan starts and an unsteady state cooling occurs until the minute 64, followed by a steady state ($t_5 \approx 42$ °C) during 6 minutes. Then the fan stops and a new NC operation happens, as follows: 6 minutes in steady state ($t_5 \approx 42$ °C), unsteady heating during 34 minutes and steady state ($t_5 \approx 78$ °C) during the next 10 minutes. At minute 120 the fan starts and a FC cooling occurs, 14 minutes in unsteady state and 6 minutes in steady state ($t_5 \approx 43$ °C). Data shown in Table 3 and Figure 7 reveal that t_5 ranges between 45 °C and 65 °C in the time intervals of 10 minutes for NC and 7 minutes for FC, respectively.

An additional set of four coupled experiments (Figure 8), where the fan started when $t_5=65$ °C and stopped when $t_5=45$ °C, was conducted in NC and FC ($n=2000$ rpm) conditions for $Q=44.8$ W. Temperature dynamics curves depicted in Figure 8 emphasize the following operation stages. The cooler works 24 minutes in NC conditions, i.e. 6 minutes in steady state ($t_5\approx 21$ °C) and 18 minutes in unsteady state, until t_5 attains a value of 65 °C. As can be seen, t_5 increases from 45 °C to 65 °C in a time interval of 10 minutes. At minute 24 the fan starts and a FC cooling takes place during the next 7 minutes, when t_5 decreases from 65 °C to 45 °C. At minute 31 the fan stops and a NC operation occurs during 16 minutes, namely 6 minutes in steady state ($t_5\approx 45$ °C) and 10 minutes in unsteady state, until t_5 increases to 65 °C. This stage is followed by a cooling of 7 minutes in FC conditions until t_5 decreases to 45 °C.

5. Conclusions

A heat pipe cooler fitted with a fins system was tested to cool an aluminium bar in order to simulate a microprocessor cooling. A data processing algorithm was developed and characteristic correlations of the process investigated were established. Experiments concerning natural and forced convection were performed and values of fins efficiency were estimated. It was concluded that the fins efficiency is heavily influenced by the flow regime, such that it becomes large at high velocity values of the cooling agent.

This developed testing procedure can be adapted to characterize any electronic device cooler. For an efficient cooling of an electronic component by means of a heat pipe system, it is important to know the system functional characteristics, which are represented by the dependencies $E=E(w, Q)$ and $\tau_s=\tau_s(w, Q)$, as shown in Figure 6. The

temperature value at the contact surface between the heat generating system and the heat spreader as a function of operation parameters is another important characteristic of the cooling system. When a temperature values range of the contact surface is imposed, the establishment of cooler fan command can be achieved using the temperature dynamics registering based on switching between natural convection and forced convection.

Nomenclature

c_a	air specific heat, $\text{Jkg}^{-1}\text{K}^{-1}$
d	distance between two fins, m
d_e	equivalent diameter of air flow cross-section, m
E	fins efficiency
FC	forced convection
G_{va}	air volumetric flow rate, m^3s^{-1}
l	fin width, m
L	fin length, m
n	speed of cooling fan rotor, rpm
N	fins number
NC	natural convection
Nu	Nusselt number
Pr	Prandtl number
q	heat flux, Wm^{-2}
Q	heat flow rate, W
Re	Reynolds number

S_b	cross-section area of aluminium bar, m^2
S_t	total transfer surface area of fins, m^2
t	temperature, $^{\circ}C$
Δt	difference between fins and environment temperature, $K (^{\circ}C)$
Δt_a	difference between air output and input temperature, $K (^{\circ}C)$
w	air velocity, ms^{-1}
z	length coordinate, m

Greek letters

α	heat transfer partial coefficient, $Wm^{-2}K^{-1}$
η_a	air viscosity, $kgm^{-1}s^{-1}$
λ	thermal conductivity, $Wm^{-1}K^{-1}$
ρ_a	air density, kgm^{-3}
τ_s	time needed to establish the steady state, min

Subscripts

a	air
b	bar
in	input
m	mean
out	output
0	initial

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Fig. 1. Schema of gravitational (left) and wick (right) heat pipe.

Fig. 2. Heat pipe system for microprocessors cooling.

Fig. 3. Thermalright SI-97 cooling system.

Fig. 4. Experimental set-up (1-heat pipes; 2-copper piece; 3-fins; 4, 5-thermocouples; 6-electric resistance; 7-aluminium bar; 8-glass cylinder; 9-glass cylinder supports; 10-alternating current variator; 11-connector; 12-reference thermocouples support; 13-set-up support; 14-data acquisition system; 15-computer).

Fig. 5. System heating dynamics ($Q=44.8$ W): **a.** natural convection; **b.** forced convection at $n=2000$ rpm ($\blacklozenge t_1, \blacklozenge t_2, \blacksquare t_3, \square t_4, + t_5, \bullet t_6, - t_7, \blacktriangle t_8, \times t_9, \circ t_{10}, \Delta t_{11}$).

Fig. 6. Fins efficiency, E ($\blacklozenge Q=34.8$ W, $\blacksquare Q=44.8$ W, $\blacktriangle Q=54.8$ W), and time needed to establish the steady state, τ_s ($\blacklozenge Q=34.8$ W, $\square Q=44.8$ W, $\Delta Q=54.8$ W), versus air velocity, w , for Thermalright SI-97 cooling system.

Fig. 7. Cooling stages in NC and FC ($n=2000$ rpm) conditions for $Q=44.8$ W

($\blacklozenge t_1, \blacklozenge t_2, \blacksquare t_3, \square t_4, * t_5, \bullet t_6, - t_7, \blacktriangle t_8, \times t_9, \circ t_{10}, \Delta t_{11}$).

Fig. 8. Temperature dynamics registering based on switching between NC and FC ($n=2000$ rpm)

conditions for $Q=44.8$ W ($\blacklozenge t_1, \blacklozenge t_2, \blacksquare t_3, \square t_4, * t_5, \bullet t_6, - t_7, \blacktriangle t_8, \times t_9, \circ t_{10}, \Delta t_{11}$).

Table 1

Steady state data for natural and forced convection ($Q=44.8$ W)

Table 2

Characteristic parameters of heat pipe cooler in steady state conditions

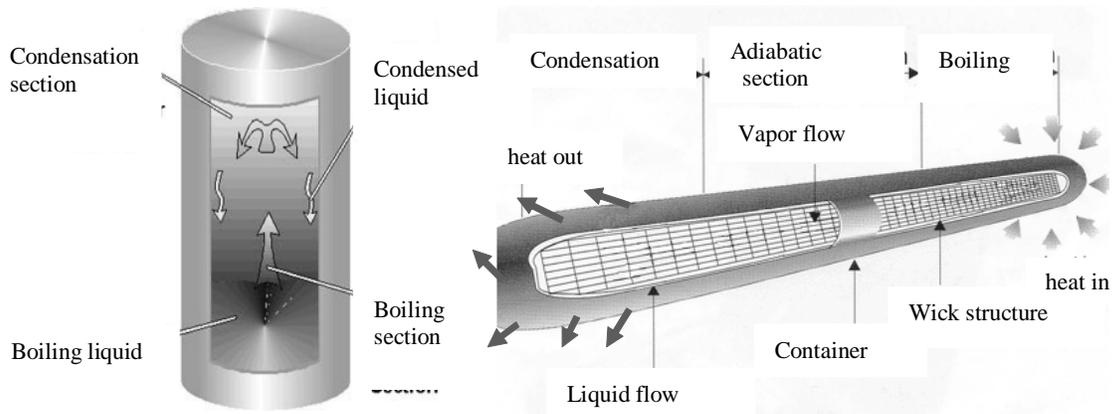
Table 3

Time variation of interface temperature ($Q=44.8$ W, $n=2000$ rpm)

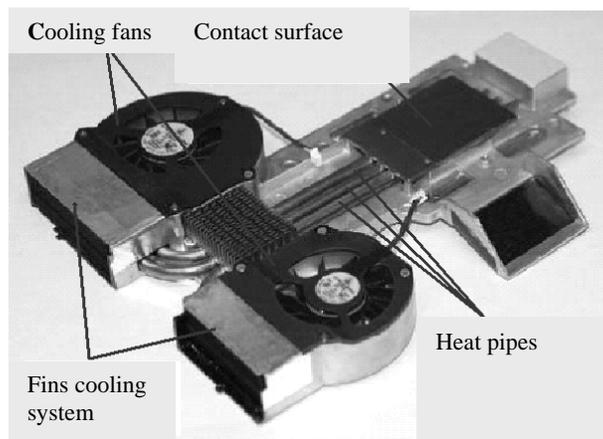
Temperature [⁰ C]	Exp 1 <i>n</i> =0 rpm natural convection	Exp 2 <i>n</i> =500 rpm 1 forced convection	Exp 3 <i>n</i> =900 rpm 2 forced convection	Exp 4 <i>n</i> =1250 rpm 3 forced convection	Exp 5 <i>n</i> =2000 rpm 4 forced convection	Exp 6 <i>n</i> =2800 rpm 5 forced convection
t ₁	89.1	63.7	63.1	62.3	61.7	61.3
t ₂	86.0	59.3	58.4	57.6	57.0	56.7
t ₃	84.3	56.3	55.2	54.5	53.9	53.5
t ₄	81.6	52.4	51.4	50.3	49.7	49.4
t ₅	78.2	49.0	47.8	46.9	46.3	46.0
t ₆	66.9	30.5	28.5	26.8	25.9	25.6
t ₇	60.1	27.3	25.6	24.1	23.1	22.7
t ₈	57.0	26.4	24.4	22.9	21.5	21.2
t ₉	66.9	28.9	26.8	25.2	24.0	23.6
t ₁₀	68.7	27.4	25.3	23.4	22.4	22.2
t ₁₁	66.3	24.4	22.6	21.2	20.5	20.2

Exp	Q [W]	n [rpm]	τ_s [min]	Δt_a [K]	Δt [K]	w [ms ⁻¹]	Re	Nu	α [Wm ⁻² K ⁻¹]	E
-	Step 1,2,4	Step 1	Step 3	Step 1	Step 5	Step 6	Step 7	Step 8	Step 9	Step 10
1	34.8	0	42	10.0	39.2	0.38	83.91	-	8.32	0.40
2		500	41	7.3	5.7	0.52	114.94	6.23	54.84	0.41
3		900	38	4.0	3.7	0.95	209.77	8.42	74.08	0.47
4		1250	27	2.6	2.2	1.45	322.72	10.44	91.89	0.64
5		2000	23	1.7	1.4	2.22	493.57	12.91	113.64	0.83
6		2800	20	1.2	1.1	3.15	699.23	15.37	135.26	0.91
7	44.8	0	40	12.0	43.1	0.41	90.02	-	8.52	0.45
8		500	39	8.3	6.2	0.59	130.15	6.63	58.35	0.46
9		900	35	5.0	4.2	0.97	216.05	8.54	75.18	0.52
10		1250	25	3.3	2.7	1.48	327.34	10.51	92.55	0.68
11		2000	20	2.1	1.6	2.32	514.40	13.18	116.01	0.89
12		2800	18	1.5	1.3	3.25	720.16	15.59	137.27	0.96
13	54.8	0	38	14.0	47.2	0.43	94.39	-	8.71	0.50
14		500	37	9.3	6.6	0.64	142.09	6.93	60.97	0.51
15		900	32	6.0	4.6	0.99	220.23	8.62	75.91	0.58
16		1250	23	4.0	3.1	1.49	330.35	10.56	92.97	0.72
17		2000	18	2.3	1.8	2.59	574.52	13.93	122.61	0.95
18		2800	16	1.7	1.5	3.50	777.29	16.20	142.61	0.99

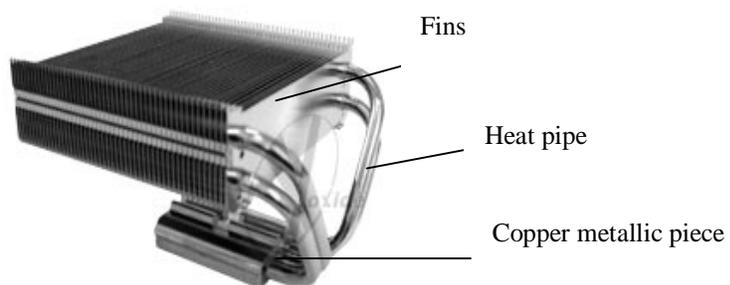
τ [min]	t_5 [°C]	State	Convection
0-6	20.7-21.0	steady	
6-14	21.0-45.0		
14-24	45.0-65.0	unsteady	natural
24-40	65.0-78.2		
40-50	78.2-78.4	steady	
50-51	78.4-65.2		
51-58	65.2-45.0	unsteady	forced
58-64	45.0-42.4		
64-70	42.4-41.7	steady	
70-76	41.7-42.1	steady	
76-77	42.1-45.3		
77-87	45.3-65.0	unsteady	natural
87-110	65.0-77.6		
110-120	77.6-79.0	steady	
120-121	79.0-65.2		
121-128	65.2-45.1	unsteady	forced
128-134	45.1-43.0		
134-140	43.0-42.5	steady	



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