Energy Conversion and Management 52 (2011) 2937–2944

Contents lists available at ScienceDirect

Energy Conversion and Management

journal homepage: www.elsevier.com/locate/enconman

Experimental analysis and FEM simulation of finned U-shape multi heat pipe for desktop PC cooling

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ARTICLE INFO

Article history:
Received 16 August 2010
Accepted 5 March 2011
Available online 7 May 2011

Keywords:
Finned heat pipe
Desktop PC-CPU
Thermal resistance
Heat transfer coefficient
Effective thermal conductivity
Forced convection

ABSTRACT

This paper presents the performance analysis of a finned U-shape heat pipe used for desktop PC-CPU cooling. The experiments are conducted by mounting the system vertically over a heat source situated inside a rectangular tunnel, and force convection is facilitated by means of a blower. The total thermal resistance (R_t) and heat transfer coefficient are estimated for both natural and forced convection modes under steady state condition, by varying the heat input from 4 W to 24 W, and the air velocity from 1 m/s to 4 m/s. The coolant velocity and heat input to achieve minimum R_t are found out and the corresponding effective thermal conductivity is calculated. The transient temperature distribution in the finned heat pipe is also observed. The experimental observations are verified by simulation using ANSYS 10. The results show that the air velocity, power input and heat pipe orientation have significant effects on the performance of finned heat pipes. As the heat input and air velocity increase, total thermal resistance decreases. The lowest value of the total thermal resistance obtained is 0.181 °C/W when heat input is 24 W and air velocity 3 m/s. The experimental and simulation results are found in good agreement.

1. Introduction

In the computer industry, the growing development and demand of processing power necessitate efficient design of processors to conduct operations faster; consequently, the need for cooling techniques to dissipate the associated heat is quiet obvious. Hence, it is highly desirable to explore high-performance cooling devices, especially for CPU cooling.

The conventional way to dissipate heat from desktop computers was forced convection using a fan with a heat sink directly. However, with the smaller CPU size and increased power as encountered in modern computers, the heat flux at the CPU has significantly increased [1]. At the same time, restrictions have been imposed on the size of heat sinks and fans, and on the noise level associated with the increased fan speed. Consequently, there has been a growing concern for improved cooling techniques that suit the modern CPU requirements. As alternatives to the conventional heat sinks, two-phase cooling devices such as heat pipe and thermosyphon, have been emerged as promising heat transfer devices with effective thermal conductivity over 200 times higher than that of copper [2]. Groll et al. [3] reported a meticulous review of the history and developments up to the year 1998, of the application of heat pipe technology for electronic cooling. Later on, Vasilev [4] provided an outline of conventional heat pipes, miniature and micro heat pipes, loop heat pipes, spaghetti heat pipes, pulsating heat pipes and some applications. Maydanik [5] reported an exclusive review on developments in loop heat pipes and their applications. Few recent experimental works on the use of heat pipes in electronic cooling include those of Naphon et al. [6], Xiaowu et al. [7], Yong et al. [8], and Liu and Zhu [9]. However, as the present study focuses on finned heat pipes for desk top PC-CPU and other large size electronic devices, the related previous works are reviewed, and presented as follows.

When fins are attached to a surface, the highest heat transfer rate will be realized if the fin base and heat source temperatures are the same. This limit is realized only for a fin base material of infinite thermal conductivity. If properly designed, heat pipes can be an effective means for bridging the thermal path between the heat source and fin base, thereby enhancing heat transfer from the fins [10]. Zhao and Avedisianz [10] performed experiments on heat transfer from an array of copper plate fins supported by a copper heat pipe and cooled by forced convection. It was demonstrated that for some conditions, fins of fixed pitch supported by a heat pipe dissipated higher heat transfer rates for the same surface temperature, than fin arrays supported by a solid rod. Legierski et al. [11] simulated cooling fins equipped with heat pipes for high power and high temperature electronic circuits and devices, and demonstrated the superiority of the proposed system over the traditional devices. Kim et al. [12] developed a cooling module in the form of remote heat exchanger using heat pipe for Pentium-IV CPU

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doi:10.1016/j.enconman.2011.03.001

as a means to ensure enhanced cooling and reduced noise level compared to the fan-assisted ordinary heat sinks. Saengchandr and Afzulpurkar [13] proposed a system that combined the advantages of heat pipe and thermoelectric modules, for desktop PCs.

Recently, heat sinks with finned U-shape heat pipes have been introduced for cooling the high-frequency microprocessors such as Intel Core 2 Duo, Intel Core 2 Quad, AMD Phenom series and AMD Athlon 64 series [14]. Wang et al. [15] have experimentally investigated the thermal resistance of a heat sink with horizontal embedded U-shape heat pipes. They showed that two heat pipes embedded in the base plate carried 36% of the total dissipated heat from CPU, while 64% of heat was delivered from the base plate to the fins. Furthermore, when the CPU power was 140 W, the total thermal resistance was at its minimum (0.27 °C/W). Subsequently, Wang [16] performed similar analysis on four heat pipes and the results were compared with those obtained by two heat pipes and without heat pipe. Using four heat pipes, the total thermal resistance could be reduced to 0.24 °C/W when the heating load was between 40 W and 240 W. Very recently, Liang and Hung [14] studied the thermal performance of a heat sink with finned U-shape heat pipes which was compatible for a wide range of high-frequency microprocessors. The optimum range of operating heat load based on thermal resistance analysis of the heat sink was characterized. The optimum L-ratio (ratio of the evaporator section length to the condenser section length) of the U-shape heat pipe was found to be dependent on heat pipe diameter and the fin spacing. Experimental observations were compared with theoretical results.

In the current study, attempt is made to extend the works of Wang et al. [15], Wang [16] and Liang and Hung [14]. In Refs. [15,16], the heat pipes were embedded horizontally inside the fins and the heat from the base was transferred simultaneously to both the heat pipes and fins. Moreover, one set of risers of the U-shape heat pipes were functioning as the evaporating section while the other set acted as condensing section. In the present study, four U-shape heat pipes are arranged vertically in such a way that the bottom acts as the evaporating section and the risers act as the condensing section. In this configuration, the heat from the base is transferred solely to the heat pipes and then to the fins. Thermal analysis is performed under both natural and forced convection modes and the results were verified by simulation using ANSYS 10 software. The transient temperature distribution in the heat pipe is also studied.

2. Materials and methods

2.1. Description of the U-shape heat pipe

The finned U-shape multi heat pipe under investigation is shown in Fig. 1. The material is the same as used by Liang and Hung [14] who used single U-shape heat pipe with circular fins; in this study, four U-shape heat pipes are used. Each condenser section is 112 mm long with 56 aluminum fins making 112 fins. Detailed views and dimensions of the heat pipe are shown in Fig. 2. The total surface area of fins is about 0.3 m^2.

2.2. The experimental setup

Fig. 3 shows the experimental setup which includes the wind tunnel, blower, flow rate controller, AC power supply, electrical heater (heat source), multi meter, desktop PC, data acquisition system (Advantech USB-4718, 8-channel) and thermocouples (k-type). The heat transfer surface of heat source is attached to the bottom of the evaporator section of the heat pipe that is vertically oriented inside the wind tunnel that has a cross section of 70 mm × 240 mm.

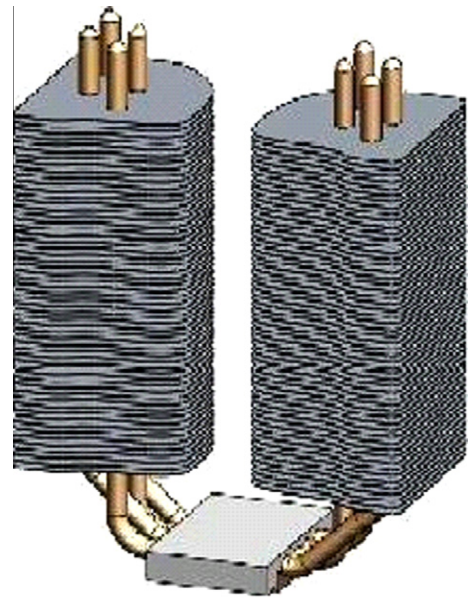

Fig. 1. Finned U-shape multi heat pipe.

As shown in Fig. 4, the thermocouples measure the temperatures at the base (T_1), evaporator (T_2), condenser lower side (T_3), condenser upper side (T_4) and the ambient (T_a). For both natural and forced convection modes, the experiment starts with a heating power of 4 W and increases it up to 24 W by increments of 4 W, each time noting the steady state (reached after one and a half hours) temperatures at the aforementioned locations. For the forced convection, the airflow velocity is varied as 1, 2, 3 and 4 m/s, by adjusting the flow control valve. The coolant velocity is measured by means of Pitot tube. In order to study the transient temperature distribution, temperatures at each 1 min from the start of the experiment, are also noted, until the steady state.

2.3. Thermal analysis

The objective of the current study is to study the total thermal resistance (R_t) and the heat transfer coefficient of the heat pipe under various operating conditions, and to verify the results by FEM simulation. Accordingly, R_t is evaluated for both natural and forced convection conditions for various heat inputs. The air velocity and heat input, that yield the minimum R_t are then found out, and the effective thermal conductivity of single finned U-shape heat pipe (k_{eff}) at this condition is calculated. The various steps to estimate R_t and k_{eff} are as follows. The thermal resistance network of the system is shown in Fig. 5.

The rate of heat transfer from the base to the surrounding air (Q) can be expressed as:

$$Q = \frac{\Delta T}{R_t}, \quad \Delta T = T_b - T_a \tag{1}$$

where T_b denotes the base temperature of the heat pipe, which is equal to T_1. The thermal resistances between the base and the evaporator (R_b), and the evaporator and the condenser (R_h) are given by:

$$R_b = \frac{T_b - T_e}{Q} \tag{2}$$

$$R_h = \frac{T_e - T_c}{Q_1} \tag{3}$$

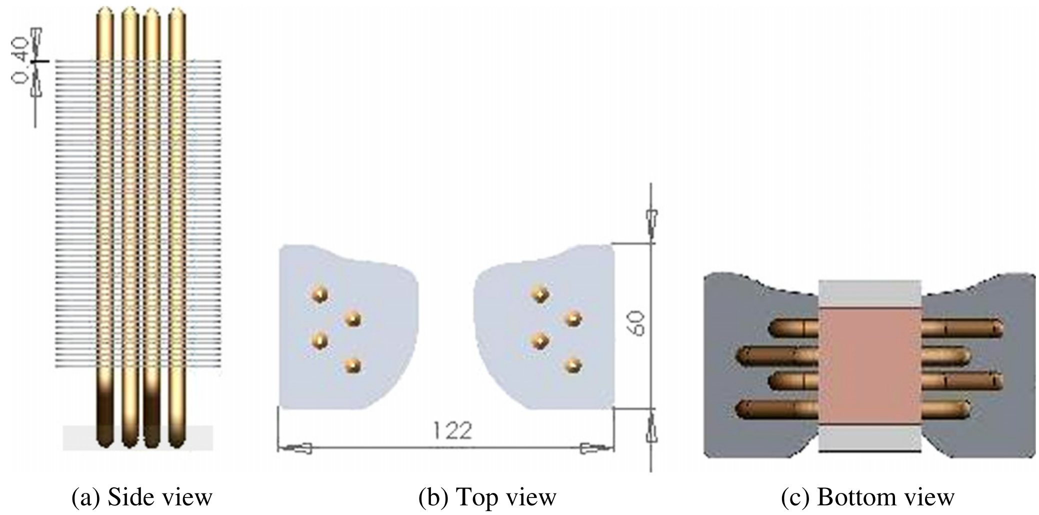

Fig. 2. Details of the finned heat pipe.

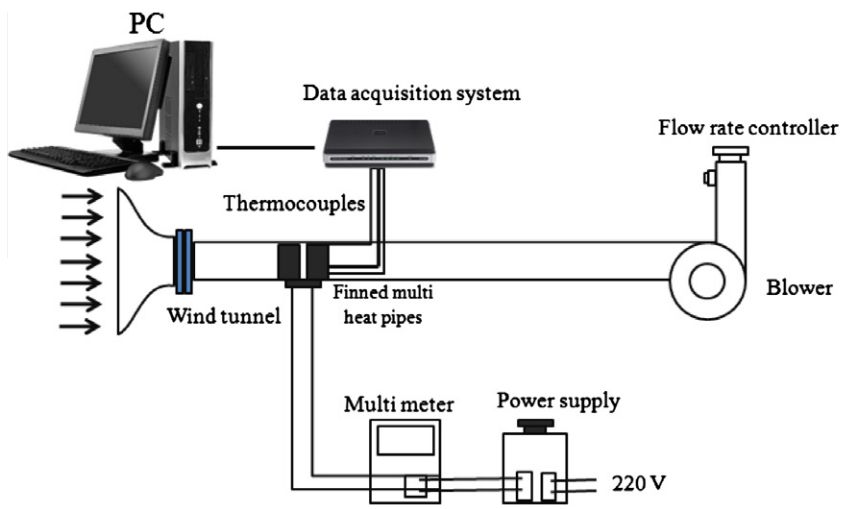

Fig. 3. Schematic diagram of experimental apparatus.

where T_e is the evaporator temperature that is equal to T_2 and T_c is the mean condenser temperature that is chosen as the arithmetic mean of T_3 and T_4. $Q_1 = \frac{Q}{a}$ according to Fig. 5, by assuming equal distribution of Q to all the eight risers. The thermal resistance of the fins (R_f) is calculated as:

$$R_f = \frac{T_c - T_a}{Q_1} \tag{4}$$

Assuming that the system is symmetric,

$$R_t = R_b + \frac{1}{\sum_{1}^{8}\left(\frac{1}{R_h + R_f}\right)} \tag{5}$$

The effective thermal conductivity of single U-shape finned heat pipe is estimated by:

$$K_{eff} = \frac{Q_h L_{eff}}{A \Delta T} \tag{6}$$

Where

$L_{eff} = (L_e + L_c)/2 + L_{ad}$.

Q_h is the heat input to one U-shape heat pipe, which is quarter of Q. L_e, L_c and L_{ad} denote the lengths of evaporator, condenser and adiabatic sections respectively of the heat pipe. A is the cross-sectional area of the heat pipe and $\Delta T = T_e - T_c$.

The heat transfer coefficients for natural (h_{nc}) and forced (h_{fc}) convections are obtained from Eqs. (7) and (8) respectively, as follows [17]:

$$Nu_{nc} = \frac{h_{nc} L_c}{k_{air}} \tag{7}$$

$$Nu_{nc} = 0.54 Ra^{\frac{1}{4}} \quad \text{and } Ra = Gr \cdot Pr = \frac{g\beta(T_{fs} - T_a)L_c}{\upsilon^2} Pr$$

$$Nu_{fc} = \frac{h_{fc} D_c}{k_{air}} \tag{8}$$

The Nusselt number (Nu_{fc}) for developing flow between isothermal parallel plates is given by [18]:

$$Nu_{fc} = 7.55 + \frac{0.024 X^{*-1.14}}{1 + 0.0358 X^{*-0.64} Pr^{0.17}}$$

where $X^* = \frac{x}{D_c \cdot R_e \cdot P_r}$ and $D_c = 2s$ for parallel plate fins, where s is the distance between two fins.

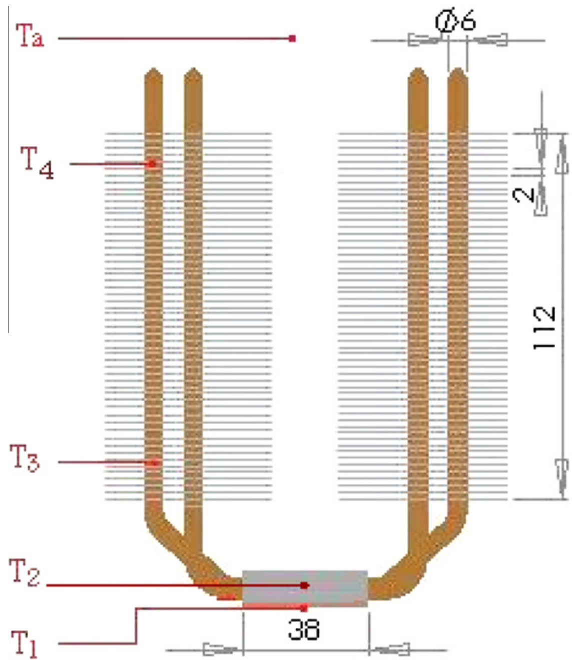

Fig. 4. Heat pipe dimensions and thermocouple locations (all dimensions in mm).

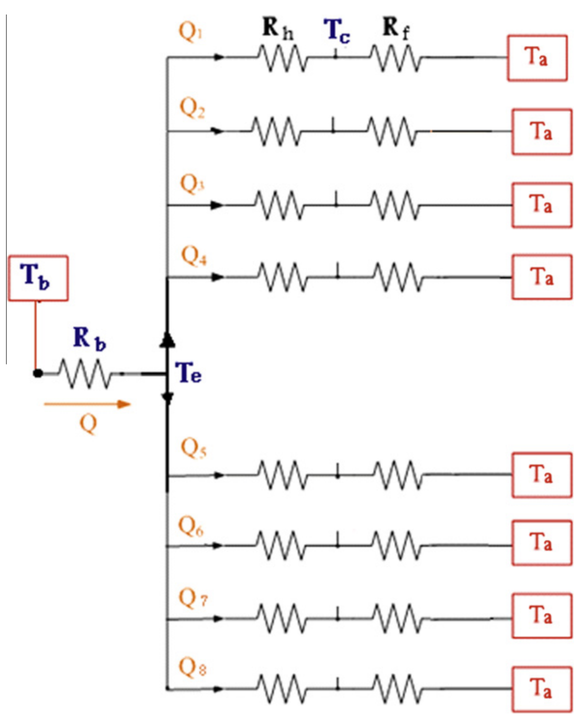

Fig. 5. Thermal resistance network.

3. Experimental results

Fig. 6 shows the total thermal resistance for different heat inputs under natural and forced convection modes. In the case of natural convection, the minimum R_t is 0.441 °C/W that is at heating load 24 W; the trend predicts further decrease in R_t at higher heat loads, as also observed by Wang [16]. With single U-shape finned heat pipe, Liang and Hung [14] obtained minimum R_t above 0.5 °C/W at around 60 W. However, when two heat pipes were sued, Wang et al. [15] could reduce R_t to 0.3 °C/W at 60 W, and

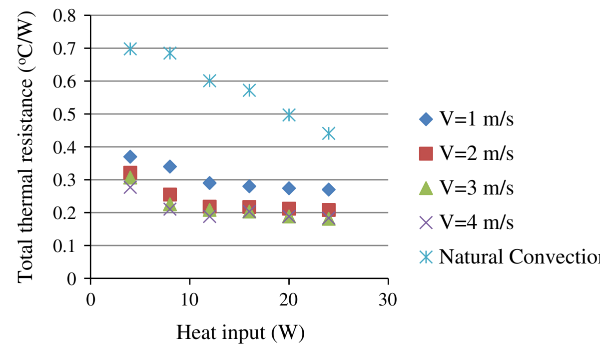

Fig. 6. Total thermal resistance vs. heat input.

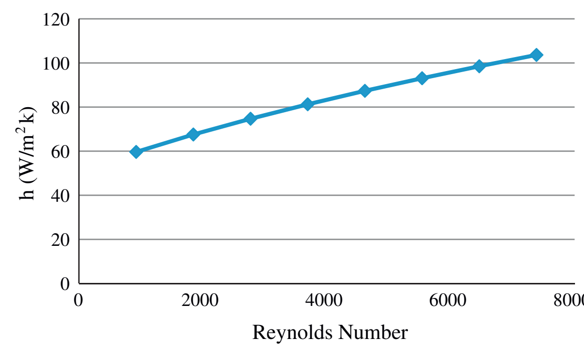

Fig. 7. Heat transfer coefficient vs. Reynolds number (velocity).

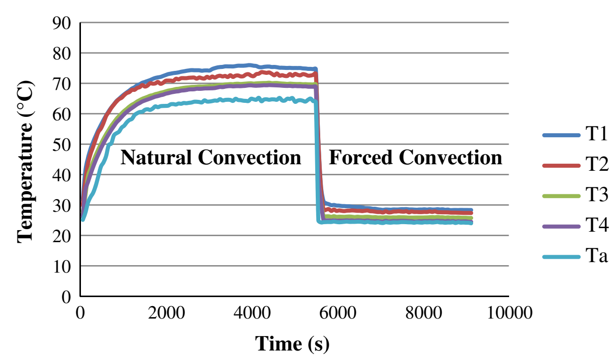

Fig. 8. Transient temperature distribution within the finned U-shape heat pipe.

interestingly, a reduction to 0.27 °C/W was possible at 140 W beyond which the twin heat pipe could not perform well. With the use of four heat pipes, Wang [16] obtained further reduction in Rt to o.24 °C/W under heating load range of 40–240 W.

Thus, it was well established that, even though the increase in heat load could decrease R_t, for a given number of embedded heat pipes, there was a limit for maximum heating load that could yield minimum R_t. Further, as the number of embedded heat pipes increased, the heat load bearing capacity of the system also increased. By comparing the present value of R_t (0.44 °C/W at 24 W) with that of Liang and Hung [14] who got more than 0.7 °C/W at 24 W (Fig. 9 of Ref. [14]) it can be deduced that the reduction in R_t is due to the increased number of U-shape heat pipes and the vertical orientation. The vertical mounting of the heat pipe is expected to enhance the performance by the accelerated condensate return flow by gravity.

Similar trend is observed in the case of forced convection as well, but significant reduction in R_t is achieved. It is seen that,

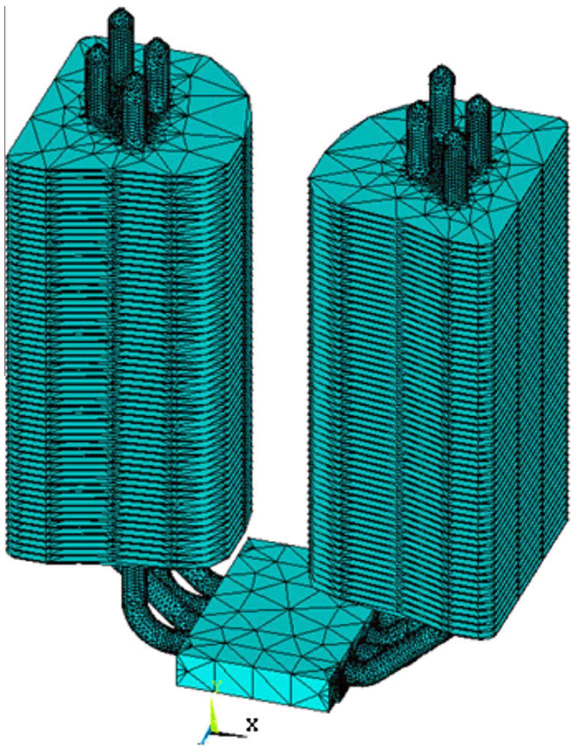

Fig. 9. The meshed simulation model.

Table 1
Material properties.

Property	Copper	Fins (Aluminum)	Heat pipe
Thermal conductivity (k) W/mK	402	186	24,000
Specific heat (C_p) J/kg K	385	895	385
Density (kg/m^3)	8933	2800	8933

the minimum R_t is 0.181 °C/W at air velocity 3 m/s, and heat load 24 W. In both the cases, the trend shows that there is enough scope for further reduction of R_t, by increasing the heat load. Further, the heat transfer coefficients h_{nc} and h_{fc} respectively for natural and forced convection, conditions are estimated. According to Eq. (7), h_{nc} is calculated to be 9.6 W/m^2 K. It is found the heat transfer coefficient increases with increase in air velocity (varied from 0.5 m/s to 4 m/s with increment of 0.5 m/s) as shown in Fig. 7. It is found that at 3 m/s (Re = 5668), h_{fc} is 93 W/m^2 K.

Fig. 8 shows the temperature history of the finned heat pipe under natural and forced convection conditions. Under the natural convection mode, the temperatures gradually increase until 2100 s (35 min) and remain almost steady thereafter. The temperature of the base reaches around 74.68 °C and the ambient temperature increases from 24 °C to 64.5 °C. After 5400 s, the system is switched over to forced convection with air velocity of 3 m/s. It is observed that the temperature at all points diminishes due to the enhanced heat removal by forced convection. The temperature of base is decreased from 74.68 °C to 28.35 °C, and the ambient temperature decreases from 64.5 °C to 24.02 °C.

4. FEM simulation

The purpose of simulation in the present study is to verify the experimental observations. Accordingly, the finned heat pipe as a whole is modeled by assuming it as a conducting medium, without taking into account the events occurring inside the heat pipe. The simulation is performed in commercial FEM software package, ANSYS, under natural and forced convection conditions. The three-dimensional (3D) model of the complex heat-pipe assembly is built in Solid Works-2010 and is exported to ANSYS 10. The copper base and the heat pipe are given fine meshing with element edge length of 1 mm, whereas the fins have coarse meshes of 1 cm edge length, forming a total of 666,812 triangular elements. The element type was chosen as 'Brick 8 node 70' out of 'Solid 70'. The 3D meshed model is shown in Fig. 9. The heat flux applied on the bottom of the base is calculated by Q/A_b, where Q is 24 W

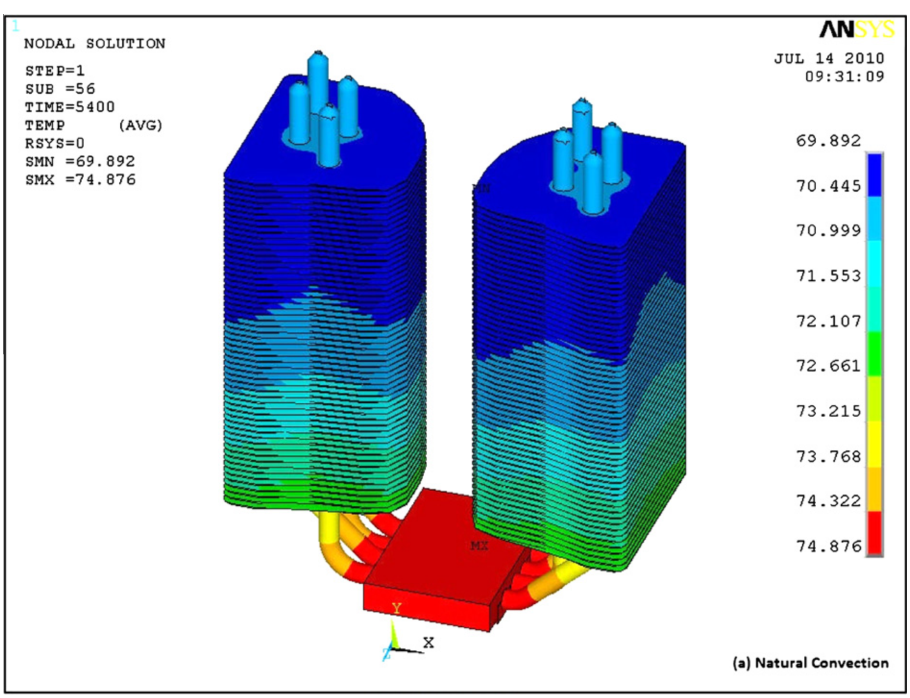

Fig. 10. Predicted temperature distribution in natural convection.

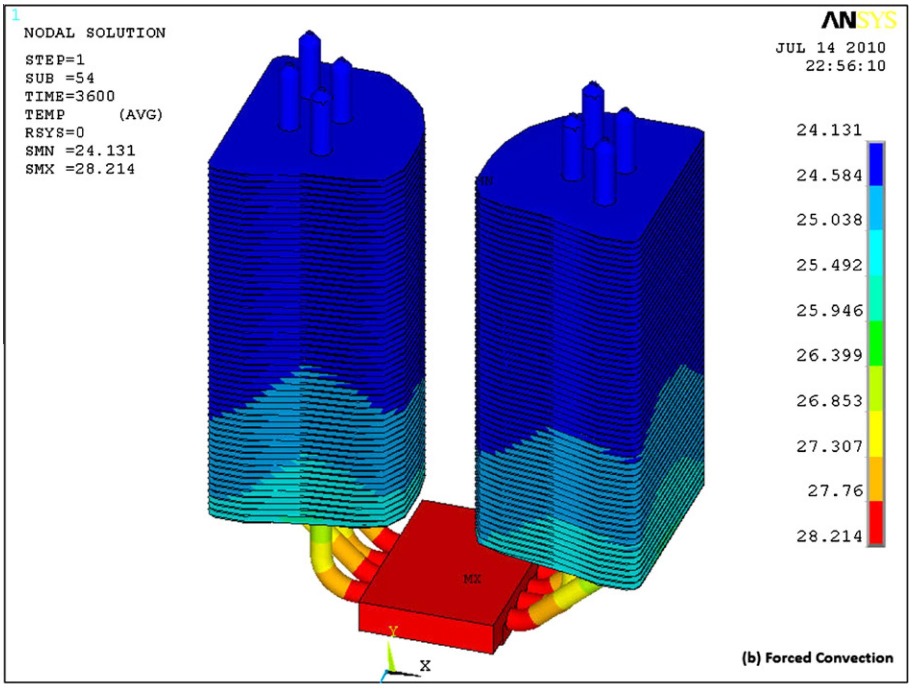

Fig. 11. Predicted temperature distribution in forced convection.

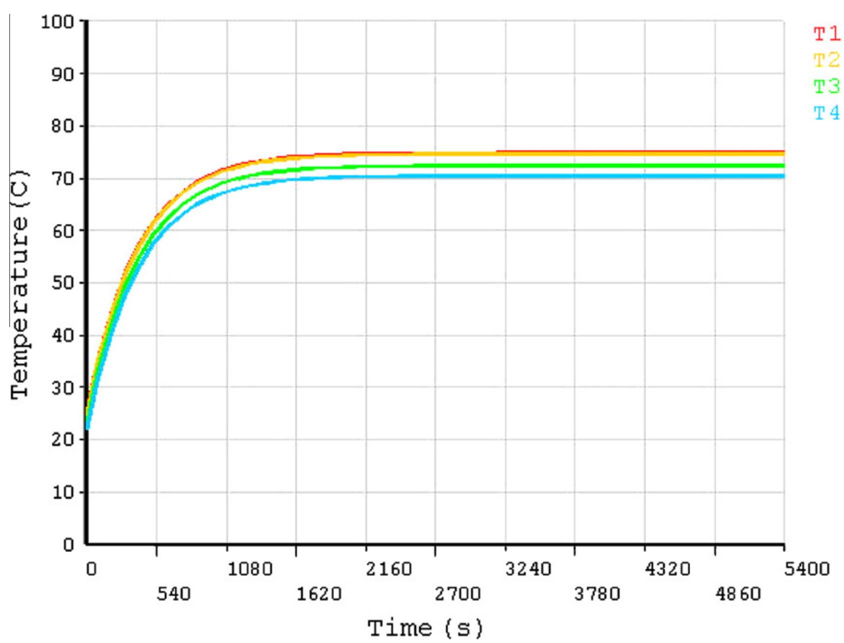

Fig. 12. Predicted transient temperature distribution at natural convection.

and A_b is the base surface area (0.00144 m^2). The initial temperature for natural convection is 22 °C, and for forced convection is 70 °C. The total time steps (thus the number of iterations) were 5400 and 3600 for natural and forced convection respectively. Thermal, h-method was opted for the analysis. The adiabatic section of the heat pipe is assumed to have no heat loss. The heat transfer coefficients h_{nc} and h_{fc} are taken as 9.6 W/m^2 K and 93 W/m^2 K (corresponding to 3 m/s) respectively. The inlet temperature of air is 24 °C. The material properties selected are summarized in Table 1. It is to be noted that the value of k for heat pipe in Table 1 is the effective thermal conductivity (k_{eff}) of single finned U-shape heat pipe, calculated using Eq. (6). The computational time was about 5 h and 3 h for natural and forced convections respectively, in a AMD Opteron 246/2 Ghz processor of 2 GB RAM and 128 MB graphic card memory.

5. Simulation results

The simulated temperature distribution in the heat pipe for natural convection at heat transfer coefficient 9.6 W/m^2 K, and heat input of 24 W, is shown in Fig. 10. After 5400 s when it comes to steady state, the highest temperature is 74.876 °C (at the base) and the lowest is 69.892 °C (top of the condenser section). Fig. 11

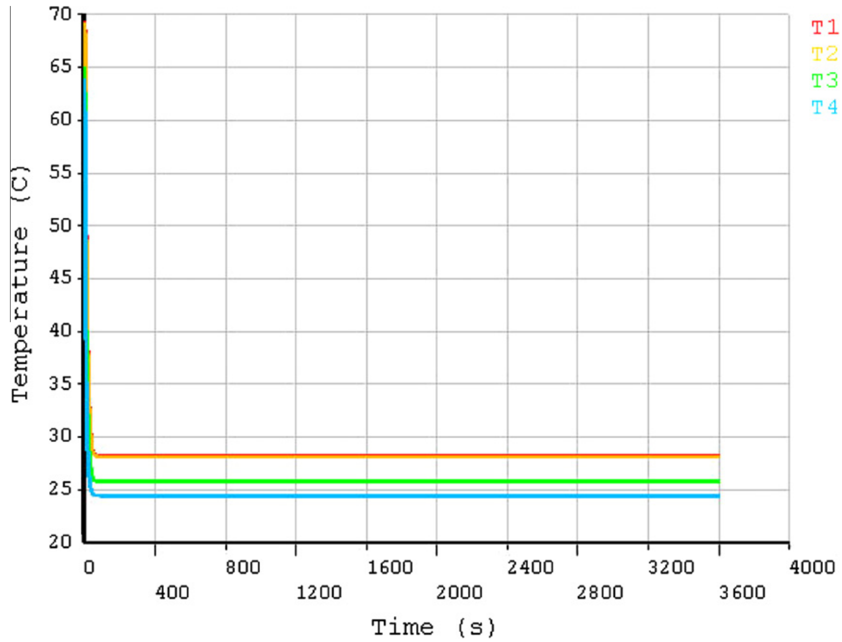

Fig. 13. Predicted transient temperature distribution at forced convection.

Table 2
Comparison of experimental and simulation temperatures.

Temperature (°C)	Natural convection		Forced convection	
	Experiment	Simulation	Experiment	Simulation
T_1	74.681	74.876	28.346	28.214
T_2	73.540	74.348	27.380	28.096
T_3	70.083	72.411	25.736	25.788
T_4	69.062	70.473	24.482	24.432

shows the predicted temperature contour for forced convection at air velocity 3 m/s and heat transfer coefficient 93 W/m^2 K. A drop in temperatures is observed, the highest being 28.214 °C and the lowest, 24.131 °C. Figs. 12 and 13 respectively show the predicted transient temperature distributions for natural and forced convection cases.

6. Comparison of experimental and simulation results

Table 2 shows the comparison of experimental and predicted temperatures at steady state, for both natural and forced convection modes. The good agreement indicates the validity of the present methodology for the thermal analysis of the heat pipe. However, a detailed simulation by incorporating the multi phase flow within the heat pipe may yield predictions that are more realistic. The predicted transient temperature distributions (Figs. 12 and 13) are also well matching with the experimentally observed trends (Fig. 8).

7. Conclusion

Experimental thermal analysis and FEM simulation of vertically oriented finned U-shape multi heat pipes are performed under natural and forced convection conditions. The results show that the air velocity and power input have important effect on the performance of finned heat pipes. The total thermal resistance decreases with increase in heat input and coolant velocity. The heat pipe orientation also plays a vital role; it is found that the vertical mounting in the present study could enhance the heat pipe performance, compared with the horizontal arrangement. The lowest value of the total thermal resistance obtained is 0.181 °C/W when the coolant velocity is 3 m/s and heat load is 24 W. Apart from the study on forced convection, the vertical mounting of the system by facilitating heat transfer directly from source to base, base to heat pipe and then to fins, is the unique feature of the current study. As the use of U-shape finned heat pipe is gaining attraction in desktop PC CPUs, the present study would definitely open ways for further research. Works are underway to study the thermal performance of the proposed system under higher heat loads.

Acknowledgement

The authors would like to thank the Ministry of Technology and Innovation, Malaysia and Universiti Sains Malaysia for the financial support for this research work.

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