Enhancement of Natural Convection Heat Transfer within Closed Enclosure Using Parallel Fins

F. A. Gdhaidh, K. Hussain, H. S. Qi

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***Abstract*—**A numerical study of natural convection heat transfer in water filled cavity has been examined in 3-Dfor single phase liquid cooling system by using an array of parallel plate fins mounted to one wall of a cavity. The heat generated by a heat source represents a computer CPU with dimensions of 37.5*37.5mm mounted on substrate. A cold plate is used as a heat sink installed on the opposite vertical end of the enclosure. The air flow inside the computer case is created by an exhaust fan. A turbulent air flow is assumed and k-ε model is applied. The fins are installed on the substrate to enhance the heat transfer. The applied power energy range used is between 15 - 40W. In order to determine the thermal behaviour of the cooling system, the effect of the heat input and the number of the parallel plate fins are investigated. The results illustrate that as the fin number increases the maximum heat source temperature decreases. However, when the fin number increases to critical value the temperature start to increase due to the fins are too closely spaced and that cause the obstruction of water flow. The introduction of parallel plate fins reduces the maximum heat source temperature by 10% compared to the case without fins. The cooling system maintains the maximum chip temperature at 64.68°C when the heat input was at 40W that is much lower than the recommended computer chips limit temperature of no more than 85°C and hence the performance of the CPU is enhanced.

***Keywords*—**Chips limit temperature, closed enclosure, natural convection, parallel plate, single phase liquid.

I. Introduction

CONTINUED miniaturizations of integrated circuits on a single computer chip have contributed to significant improvements in the performance of computer systems. As a result, the demands on power dissipation and complicated thermal control strategy have increased. The principal objective of the design of the electronic devices is to maintain components at or below specified maximum service temperatures [1]. The recommended electronic chip temperature in most applications should not exceed 85°C [2]. Natural convection heat transfer is the favoured mode for electronic cooling and considered an effective and economical cooling strategy because there is no need for any moving parts,only the buoyancy force due to temperature difference acts on the fluid causing the cooling process [3].

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Current desktop computers use direct or indirect air/liquid cooling to remove heat from the computer chip surface [4], [5]. Due to the inherent heat transfer limitations of air cooling, a distribution system consisting of a pump and associated plumbing fixtures for indirect liquid cooling and undesirable noise and vibration may accompany fan or pump assemblies may be precluded for future desktop computers [6]. Therefore, there is a promising alternative way to keep the electronic devices at the safe operation temperature by using liquids. This is justified by the fact that the liquids have relatively high removal of heat when compared to air. To surpass these limits in a passive cooling system, the enhancement of heat transfer is necessary [1].

In natural convection, the fluid motion inside enclosures whether these enclosures were vertical or horizontal occurs due to the temperature gradient change, therefore the isothermal boundary condition is adopted [7]–[9]. A comprehensive review of convection cooling options in electronic packages is provided by [4], [10]. Parallel plate fin arrays are often used to enhance heat transfer by free convection and the fluid flow between the fins is usually assumed as two-dimensional [11]. Elenbaas [12] noticed that there is an interaction between the fin height and spacing. He concluded that, if the plates are close enough to allow interaction of the adjoining boundary layers, heat transfer rates diminish as fin plates spacing is decreased. Bar-Cohen [13] analysed an array of longitudinal fins to determine optimal spacing and thickness corresponding to the maximum heat dissipation. Bar-Cohen and Jelinek [14] determined that the fin spacing should equal the fin thickness for an optimum least material fin. Heindel et al. [15] examined numerical and experimental studies to find the optimum design for parallel plate fin arrays. The purpose of their study was to predict the heat transfer from discrete heat sources mounted to one wall of a cavity and the cooled wall kept isothermally at 15°C. The surface of each discrete heat source was augmented by dense arrays of parallel plate fins. Their results showed that using fins on the heaters enhanced the heat transfer as much as 24 and 15 times for vertical and horizontal cavity orientations respectively. Gdhaidh et al. [16] studied numerically discrete heated cavity filled with dielectric liquid FC-77 and air. Although their results based on an isothermal boundary condition for the cold wall, they revealed that that in real applications keeping a wall at low constant temperature is unrealistic. Gdhaidh et al. [17] studied numerically a desktop computer heat transfer characteristics by using one phase water cooling system where the cold wall is assumed as a heat sink with air flow created by exhaust fan.

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This paper numerically investigates the effect of adding parallel plate fins on the maximum heat source temperature and also on thermal behaviour. Water was used as the heat transfer medium to remove the excessive heat generated by the electronic chips, adopting a realistic boundary conditions for the cold wall by assuming heat sink with air flow created by an exhaust fan. The optimum fin number and spacing between fins were examined to obtain the lowest maximum temperature.

II. Numerical Solution

A. Problem Description

A 3-D numerical prediction of coupling between natural convection heat transfer inside enclosure has been investigated. The air flow within the computer case is assumed turbulent and the k-εmodel was used.

Fig. 1 shows a schematic of the geometry, and Table I lists the dimensions for the simulated model. One heat source of size 37.5∗37.5mm is mounted on a vertical substrate.

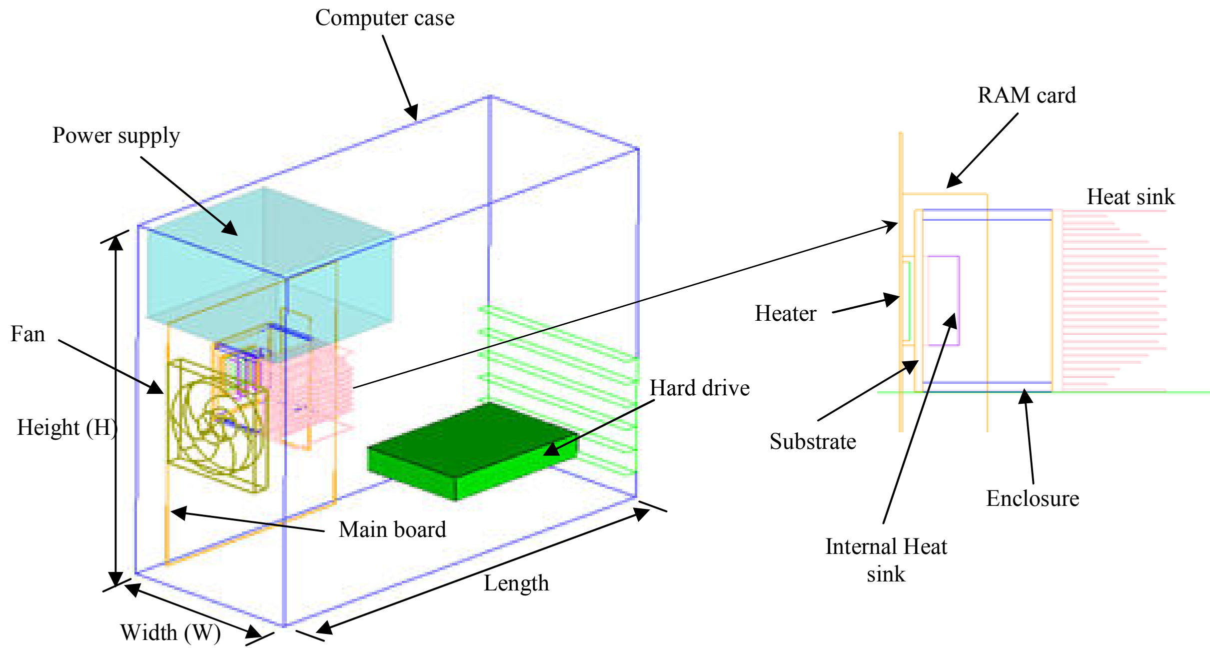

Fig. 1 Schematic diagram of the numerical physical model

TABLEI
DIMENSIONS OF THE MODEL IN mm

Components	Computer case	Hard drive	Heater	Enclosure	substrate	Main board	Power supply
Height	352	23	37.5	77	77	245	89
Length	418	145	37.5	68	68	200	138
Width	171	100	5	50	3	1	148

A dense parallel plate fin arrays were attached on the substrate at the opposite side of the heater. A rectangular enclosure filled with water located between the substrate at one end, and a heat sink without fans at the other end. The size of enclosure and other specifications are taken from previous study by [17]. An exhaust fan with 90mm diameter is installed at the end of the computer covering case to create the air flow. The applied heat input used was in the range of 15 - 40 Wand the value of power remains constant during the simulation.

For the parallel plate fins which were installed on the substrate, various fin numbers were used in order to find the optimum number of fins which give the lowest maximum temperature. Fig. 2 shows the geometry of the heat sink where the heat sink base thickness is β_{fin}, fin thickness ofδ_{fin} and fin length of L_{fin}. These were fixed at 2mm, 0.5mm and 12mm respectively as recommended by [15]. The fin base dimension is the same as the heater dimension (37.5∗37.5mm).

B. Governing Equations

Two different fluids were used in the cooling system, water in the enclosure and air within computer case. The water flow is assumed laminar and is created by the difference in the density where the Boussinesq approximation is applied while the air flow is turbulent and the k-εmodel is used. Moreover, the mathematical model was constrained by the following assumptions: 3-D steady state heat transfer, contact resistance between any solid contact surfaces assumed equal to 0.18°C/W [17]. Viscous heat dissipation and the radiation effects are neglected and an incompressible fluid flow was assumed.

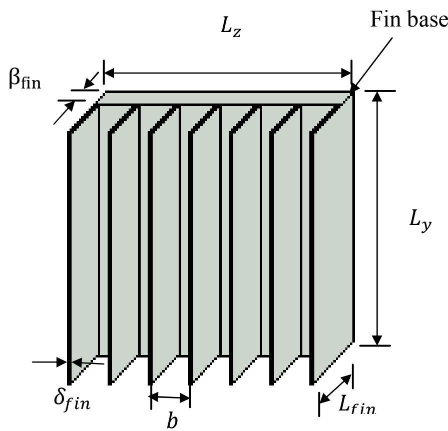

Fig. 2 Schematic diagram of the internal heat sink

Applying the above assumptions on the governing differential equations (Navier –stokes equations) of continuity, momentum and energy, yield the following equations:

Fluid Region:

Continuity equation:

$$\nabla . V = 0 \tag{1}$$

Momentum equation:

$$\rho(V.\nabla V) = -\nabla P + \mu \nabla^2 V + F \tag{2}$$

Energy equation:

$$V.\nabla T = \alpha \nabla^2 T + S \tag{3}$$

Solid Region:

In this region, only the energy equation is used as the velocity vectors u, v and w are zero. Therefore, the energy equation could be written as:

$$\nabla^2 T + \frac{S}{K_s} = 0 \tag{4}$$

where V is the velocity components, P is the pressure,ρ is the density,μ is the viscosity, T is the temperature, α is the thermal diffusivity, K_s is thermal conductivity of solid and F and S are the body force and the source term respectively.

The following boundary and initial conditions are used in order to complete the mathematical model: the flow rate inside the computer case was assumed fixed at 0.026 m^3/s. The air inlet temperature is set at 22°C, air physical properties are calculated at the same temperature and the heat energy remains constant for every test.

In the current study the ratio between the fin length to fin spacing is large (L_{fin}/ b>>1), hence, the fin edges effects are assumed negligible. If the plates are close enough to allow interaction of the adjoining boundary layers, then the average Nusselt number can be obtained by using the following expression which was developed by [15]:

$$\overline{N_{ub}} = \frac{1}{24} R_{ab} \left(\frac{b}{L_y}\right) \left\{1 - exp\left[\frac{-35}{R_a\left(\frac{b}{L_y}\right)}\right]\right\}^{3/4} \tag{5}$$

In (5), the average Nusselt number and Rayleigh number are based on the plate spacing b where:

$$R_{ab} = \frac{g\beta b^3 (T_h - T_c) P_r}{\upsilon^2} \tag{6}$$

whereβ is the volumetric thermal expansion coefficient,υ is the kinematic viscosity, T_h is the hot wall temperature, T_c is the cold wall temperature, g is the gravity, P_r is Prandtl number and b is the fin spacing.

C. Solution Procedure

A CFD software was used based on a Finite-Volume scheme to convert the governing equations to algebraic equations that can be solved numerically. The “SIMPLE” algorithm (semi-implicit method for pressure linked equations) is used to handle the coupling between pressure and velocity. Moreover, to avoid the divergence in the iteration process, Under-relaxation techniques are used to slow down the changes between iterations.

The solutions are deemed to be converged when the residual of 10^{-5} for the equations of continuity and momentum and 10^{-7} for the energy and k-εmodel equations are reached. Furthermore, the effects of the elements number on the results are examined. The final optimised number of elements is 416,640 when the internal heat sink on the substrate has 18 fins and that gave stable solution respecting to the accuracy and running time.

III. Results and Discussion

A numerical study was carried out to examine the effect of addition a heat sink mounted on substrate inside closed enclosure. During the design of the cooling system, some dimensions have been taken from previous studies of [15], [17]. The simulated working fluid inside the enclosure is water and the corresponding Prandtl number P_r is assumed to be 5. Moreover, different numbers of fins were tested and the applied heat input was set at 40W to find the lowest value of maximum system temperature.

The effect of using different heat input level (Q) in the range of 15 - 40W was also examined and then the results are compared with the case without using internal heat sink. Normally when two surfaces are in contact, less than 1% of the surfaces are in touch, due to roughness [18]. Therefore applying thermal paste with high thermal conductivity among contact surfaces reduces the value of thermal resistance between these surfaces. As a result, a thermal resistance value of 0.18°C/W was assumed between any two solid surfaces in contact.

A. Effect of Fin Numbers (n)

In this study, the parallel plate fin arrays are designed to obtain the maximum power dissipation. The fin material was selected as aluminium because of its high thermal conductivity

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(205W/mK), structural strength and durability. The heat sink base has the same dimensions of the heater and attached to the substrate at the opposite side of the heater. Because the water flow inside the enclosure is created by buoyancy force in Y direction, the parallel plates (fins) direction is also in Y direction to prevent the obstruction of water flow. Fig. 3 illustrates the fitted curve of the fin number effect on the maximum system temperature when 40W heat energy was used.

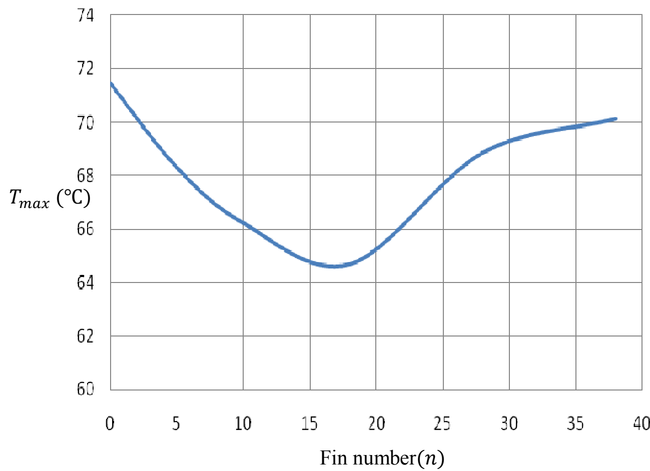

Fig. 3 Fin number against maximum system temperature for a parallel plate fin array with a heated base area of 37.5*37.5mm and 40W heat input

As the number of fins increase, the maximum temperature went down to 64.68°C when the heat sink had 18 fins. After that, point when the number of fins increases the maximum temperature also increased. In addition the maximum temperature went down by 10 % when 18 fins heat sink was used compared to the case without parallel plate fin.

The following equation was used to obtain the fin spacing b that is dependent on the number of fins (n):

$$b = \frac{L_z-(n*\delta_{fin})}{(n-1)} \quad (7)$$

For array of parallel plate fins, the fin spacing may not correspond to the maximum total power dissipation because the fins are too closely spaced. Therefore, heat transfer from each fin is significantly reduced and total power dissipation from the array may decrease, although more surface area is added. In general the fluid motion is weak due to the natural convection which creates the flow as a result of temperature differences and when the spacing between adjacent fins become too narrow due to high number of fins the flow is blocked. Therefore, an optimum geometry was obtained to maximize total power dissipation from a fin array. Using (7) the optimised fin spacing was obtained and equal to 1.7mm.

B. Heat Input Effects

In the present investigation, the heat input used varied between 15 - 40W. The heater material used which produces constant heat input is the extruded aluminium with thermal conductivity of (k_h=205W/mK) and the substrate material chosen was copper with thermal conductivity (k_s=387W/mK). The fluid Prandtl number of water is assumed to be fixed at 5. Also the fluids and the material properties were assumed constant. The effects of different values of heat energies on the maximum temperature and average Nusselt number are presented.

1. Maximum System Temperature

Fig. 4 shows the comparison of the maximum system temperature which occurs in the heater between the present study and the work of [17] with different values of heat input (Q).

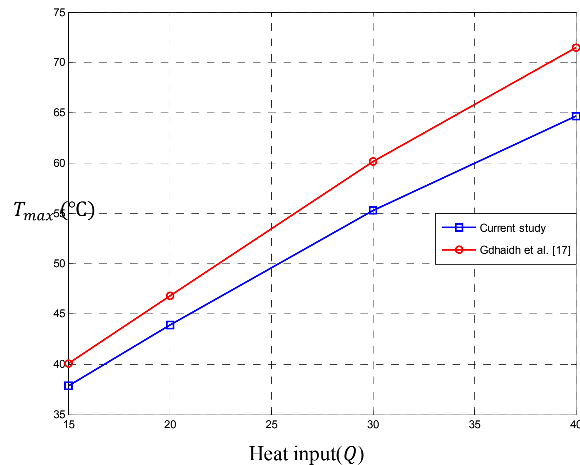

Fig. 4 Comparison between the maximum system temperature with different heat energy input for the present study and [17]

It was found that the maximum temperature of the system increases as the heat input increased. Moreover, the temperature curve is almost linear which indicates that the temperature only depends on the heat input when the other conditions were fixed. By comparing the two results, the current study has better cooling results than that presented in [17] when a heat sink was attached to the substrate where the heat dissipation is enhanced.

At15W the maximum temperature in the present study is 5.4% lower than those presented in [17]. At 40W, however the percentage reached a value of 10%. This is mainly due to more heat transfer area is added by installing the internal heat sink with 18 fins. Another reason is that the increase in buoyancy driven flow inside the enclosure results in a decrease of the thermal boundary layer and an increase of the average heat transfer coefficient between the parallel fins.

Fig. 5 shows the temperature distribution within the enclosure, substrate and heater for two different values of heat input (15 and 40W). The maximum temperatures are 37.87°C at 15W and 64.68°C at 40W respectively.

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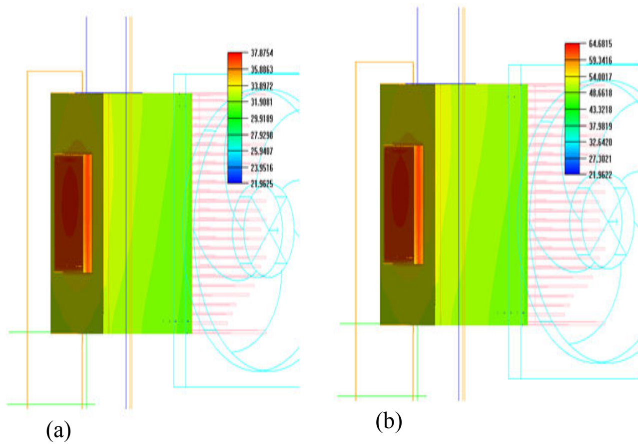

Fig. 5 Temperature distribution within enclosure, substrate and heater for (a) 15W and (b) 40W

The contour lines of temperature as shown in Fig. 5 are approximately vertical in the fluid region which indicates that, the heat transfer inside the enclosure is controlled by conduction due to the slow water motion within the enclosure. The gradient increases inside the enclosure at 40W due to the increase of the buoyant force. In addition, the maximum temperature is located almost at the centre of the heater as shown in the figure.

Fig. 6 shows the substrate/water interface temperature for different values of the applied heat input.

The temperature goes up from the base of the enclosure to reach a maximum value at the centre of the substrate due to the increase in the thermal boundary layer. At the centre, the flat part of the curve denotes zero temperature gradient because of the higher heater and substrate thermal conductivity. After this region, the temperature decreases due to the decrease in the thermal boundary layer.

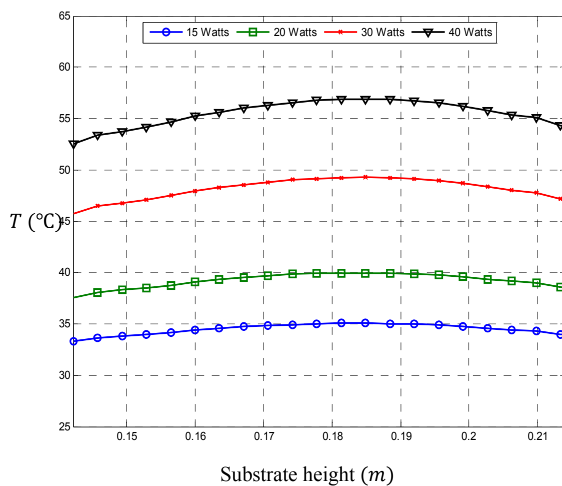

Fig. 6 The local substrate/water interface temperature with different applied heat input

The increase in the thermal boundary layer is caused by the reduction in the convection coefficient and the increase in the local bulk fluid temperature. The effects of the enclosure walls thickness are clearly shown at the substrate interior corners having steeper gradients due to the high thermal conductivity for both enclosure and substrate.

2. Average Nusselt Number

Nusselt number presents the convection and conduction heat transfer ratio, and hence a larger Nusselt number corresponds to a more active convection. Fig. 7 illustrates the mean Nusselt number for different values of heat input (using (5)). From the figure, the average Nusselt number rises as the heat input increases because of the increase in buoyancy driven flow inside the enclosure.

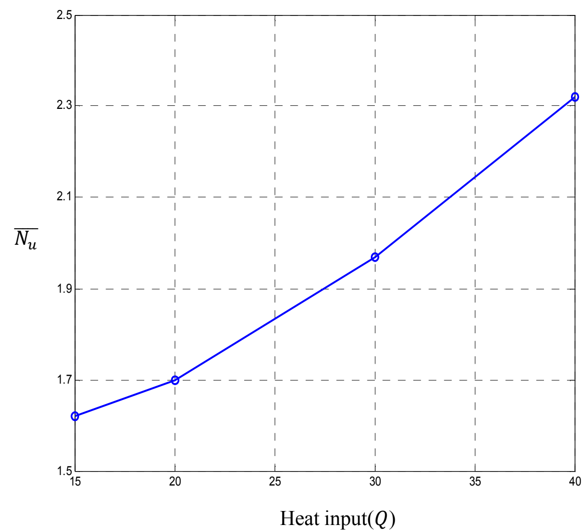

Fig. 7 Average Nusselt number with different values of heat input

As a result, the thermal boundary layer decreases and the average heat transfer coefficient increases. Moreover, the values of average Nusselt number are weak due to introducing the parallel plates inside the enclosure which blocks the water flow which is already weak. On other hand these plates increase the heat transfer area and then more heat energy dissipated to the fluid, as a result the maximum temperature reduced.

IV. Conclusions

A numerical study of a steady state 3-D natural conjugate convection analysis in a rectangular enclosure and forced convection within a computer case has been carried out. One can conclude from the results of this research the following:

- The number of parallel plates attached to the substrate has a significant effect on the value of the maximum temperature. It is also found that the maximum temperatures of the system reduced by increasing the number of fins. After certain number of plates (18) the temperature increased due to the fins are too closely spaced where the water flow is blocked and heat transfer from each fin is considerably reduced.

- The temperature at substrate/water interface in the place corresponding to the heater remains constant due to the high thermal conductivity of the heater and the substrate.
- The value of mean Nusselt number increases as the heat input increases from 15W to 40W. That is due to the thermal boundary layer being thin and the increase of the average convection coefficient. In general, the value of mean Nusselt number is not large because the fins affect the flow.
- It is found that the addition of parallel plates in the cooling system reduces the maximum temperature by 10% relative to un-finned system.
- The system with the parallel fins reduces the maximum chip temperature to 64.68°C which is much below the recommended manufacture temperature for the desktop computers.

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