Mansoura Engineering Journal, (MEJ), Vol. 23, No. 4, December 1998.

EFFECT OF HEATING ELEMENTS POSITION ON FREE CONVECTION IN A VERTICAL RECTANGULAR ENCLOSURE

تأثير موضع المسخنات الحرارية على انتقال الحرارة بالحمل الحر فى حيز رأسى مستطيل

F. F. Araid
Mechanical Power Engineering Department
Mansoura University, Egypt

خلاصـــــــة

يقدم هذا البحث دراسة تجريبية لانتقال الحرارة بالحمل الطبيعى فى حيز هوائى رأسى مستطيل ثنائى البعد. تمت الدراسة لسريان هواء رقائقى فى الحيز الهوائى عند ثبوت الفيض الحرارى بأستخدام مسخنين مثبتين على السطح الرأسى المعزول للحيز الهوائى والمواجه لسطح ذو درجة حرارة ثابتة. السطحين الافقيين العلوى والسفلى معزولين. وابعاد الحيز الهوائى ٥٠× ١٠٠× ٣٠٠ مم وذو نسبة باعية =٢ . وقد اجريت التجارب فى مدى تغير رقم رايلى من ٤,٣٣ × $١٠^{٤}$ الى ٦,٢٩ × $١٠^{٥}$ وذلك فى ثلاث حالات لاوضاع المسخنين. وقد تركزت الدراسة فى تأثير وضع المسخنين والمسافة بينهما وتغير رقم رايلى على درجة حرارة المسخنين القصوى والمتوسطة وكذلك رقم نوسلت الموضعى والمتوسط. وقد اثبتت الدراسة ان وضع المسخن السفلى بالقرب من القاعدة السفلية للحيز الهوائى والمسخن العلوى فى منتصف المسافة بين المسخن السفلى والسطح العلوى يعطى زيادة ملحوظة فى معدل انتقال الحرارة بالمقارنة بالحالات الاخرى. تم استنتاج بعض العلاقات اللابعدية بين عدد نوسلت المتوسط وعدد رايلى لحساب معامل انتقال الحرارة بالحمل الطبيعى.

ABSTRACT

Free convection in heated two-dimensional rectangular vertical enclosure is investigated experimentally for different two heater configurations. The study is carried out for the steady laminar flow of air in an enclosure with constant heat flux two heaters at one adiabatic insulated vertical wall, an isothermally cooled facing vertical wall and insulated horizontal walls The enclosure is of 50x100x300 mm width, height and spanwise dimensions respectively with an aspect ratio $A=2$. The parameters considered are $4.33\times10^4 \leq Ra \leq 6.29\times10^5$ and $Pr = 0.7$ Three different cases for the position of the two heaters are considered. The work concentrates on the effect of the position of the heat sources, the wake distance between them, and Rayleigh number on the heat sources maximum and local surface temperatures (T_{max}, $\bar{T}$) and the local and average Nusselt number (Nu, $\overline{Nu}$), which are considered as the design parameters of any electronic equipment To get minimum T_{max} and $\bar{T}$ and maximum Nu, and $\overline{Nu}$ for the upper and lower heat sources, the position of the lower heater must be nearer to the lower horizontal wall and the position of the upper heater must be in the middle of the distance between the lower heater and the upper horizontal wall. The results also show the increase of T_{max}, $\bar{T}$, Nu, and $\overline{Nu}$ for the upper and lower heat sources with the increase of Rayleigh number. T_{max} and $\overline{Nu}$ are also correlated with Rayleigh number

Accepted December 12, 1998

INTRODUCTION

An area in which has seen a considerable amount of research activity in the recent years is that of heat removal from electronic circuitry. With continued effort to decrease the size of electronic equipment, the energy dissipated per unit area has increased substantially in most engineering applications. Since the performance of electronic components is often strongly temperature dependent, there is a growing need to develop suitable methods for the cooling of electronic packages that are employed in a wide variety of applications, such as air space travel systems, communication systems, naval systems and computers. Heat removal from electronic devices usually imposes the major constraint on further reduction in size.

Previously, many investigators studied the various modes of heat transfer and relevant configurations along with the associated heat transfer. Extensive surveys of this field have been presented by Jaluria [1], Incropera [2], Papanicolaou and Jaluria [3] and El Kady and Araid [4].

Many electronic instruments, devices and systems are cooled only by natural convection, making it important to study the relevant mechanism underlying thermal transport. Jaluria [5] studied the buoyancy-induced flow due to thermal sources on a vertical isolated plate. The natural convection between series of vertical plate channels with embedded line heat sources has been studied in order to meet the relatively lower operating temperature requirements of integrated circuits by Anand et al [6], and Kim et al. [7, 8]. Another important configuration involves complete and partial enclosures. Such as those that are encountered in small electronic devices and personal computers. Not much work has been done on such enclosure flows. But there is a growing interest in these problems. Keyhani et al. [9] investigated experimentally the natural convection flow and heat transfer characteristics of an array of discrete heat sources in enclosures and in [10] they showed the aspect ratio effect on natural convection. Carmona and Keyhani [11] showed the cavity width effect on cooling of five heaters on one vertical wall of an enclosure. El Kady and Araid [4] showed the effect of size and location of a surface heater embedded in the vertical wall of a two dimensional rectangular enclosure.

From the surveyed literature it is found that, there is a lack of experimental knowledge and data of the natural convection from heat sources in the enclosures of the electronic equipment. The aim of the present work is to present experimental laminar heat transfer data of two dimensional rectangular enclosures, which have two heat sources on one vertical side wall. It concentrates on the effect of the position of the heat sources, wake distance and Rayleigh number on the surface and maximum temperatures of the heat sources as well as the local and average heat transfer from them.

EXPERIMENTAL WORK

The schematic diagram of the constructed test apparatus is shown in Fig. 1. A two dimensional rectangular enclosure is used with dimensions of height $H=100$ mm and width $W=50$ mm and 300 mm in the spanwise direction. The upper and lower

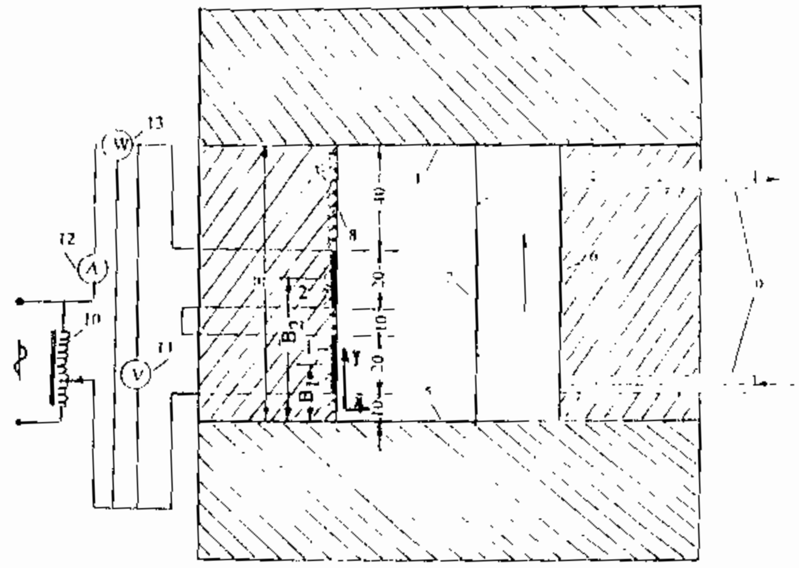

Fig. 1.a Experimental test apparatus for case (1) $B_1/H = 0.2$, $B_2/H = 0.5$

(1) and (2) electric heaters, (3) copper-constantan thermocouples, (4) lower horizontal wall, (5) upper horizontal wall, (6) heat exchanger, (7) and (8) vertical walls, (9) cooling water tubes, (10) auto-transformer, (11) voltmeter, (12) ammeter, (13) wattmeter

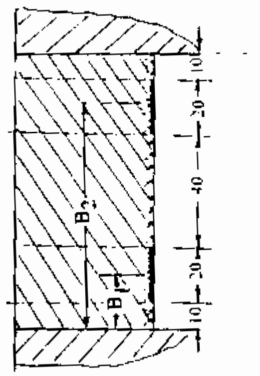

Fig. 1.b Vertical wall with heating elements for case (2) $B_1/H = 0.2$, $B_2/H = 0.8$

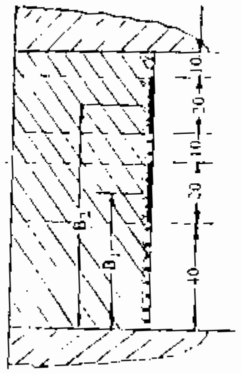

Fig. 1.c Vertical wall with heating elements for case (1) $B_1/H = 0.5$, $B_2/H = 0.8$

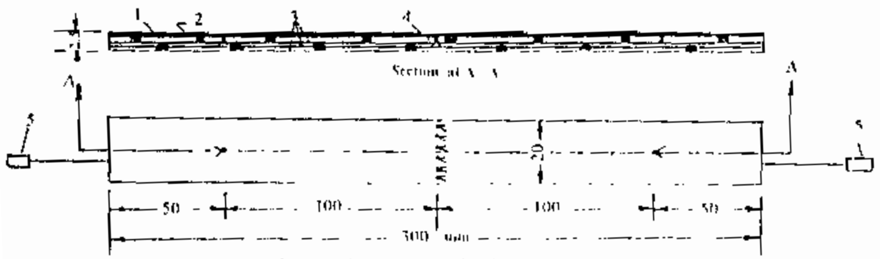

Fig. 2 Construction of heating element

(1) stainless steel sheet, (2) Ni-Cr electric heater, (3) Mica sheets, (4) copper-constantan thermocouple, (5) electric connections

horizontal surfaces of the enclosure(4) and (5) were maintained at constant temperature by constructing them from thermal insulating material (expanded polystyrene) of 50 mm thickness. The right vertical wall of the enclosure (7) was maintained approximately at constant temperature by fitting a copper counter-flow heat exchanger (6) in which the city water was used as a coolant liquid. The left vertical wall of the enclosure was made of the closed-pore extruded polystyrene insulation (8) of 50 mm thickness. Two electric heating elements (1) and (2) with dimensions of 20 mm height, 2.5 mm thickness, and 300 mm long was embedded in the left vertical adiabatic wall to make the heater face in the same vertical inside plane of the wall. The study is carried out for three cases of the two heaters positions. The left vertical wall with the two heaters positions for the three cases is shown in Figs (1-a) to (1-c). In the first case which is shown in Fig. (1-a), the leading edge of the lower heater is 10 mm height above the horizontal lower wall and the distance between the upper and lower heaters is 10 mm; $B_1/H=0.2$ and $B_2/H= 0.5$ where B_1 and B_2 are the heights of the lower heater center and the upper heater center respectively. In the second case which is shown in Fig. (1-b), the leading edge of the lower heater is 10 mm height above the horizontal lower wall and the distance between the upper and lower heaters is 40 mm; $B_1/H=0.2$ and $B_2/H= 0.8$. In the third case which is shown in Fig. (1-c), the leading edge of the lower heater is 40 mm height above the horizontal lower wall and the distance between the upper and lower heaters is 10 mm; $B_1/H=0.5$ and $B_2/H= 0.8$.

The heat input to the heating elements was controlled by an auto-transformer (10) as well as one voltmeter (11) and an ammeter (12). The heat input to the heating elements was also checked using a wattmeter (13). Once the power switch closed, the heating system starts

The heating element construction is shown in Fig. 2. The heating element face is made of polished stainless steel sheet (1) with 0.5 mm thickness. This sheet metal was heated electrically by an electric nickel-chromium heater (2) with rectangular cross section of 0.25x0.5 mm and an electric resistance of 8.3 Ohm/meter length. The electric heater which has total resistance of about 50 Ohm is wounded around a mica sheet of 0.5 mm thickness (3).

The temperatures of each heating element are measured by six copper-constantan thermocouples of 0.3 mm wire diameter and fixed on the back surface of the sheet metal. The positions of the six thermocouples on the sheet metal are shown in Figs. 1 and 2. Four thermocouples are fixed in the middle vertical plane of the spanwise as shown in Fig. 2. The other two thermocouples are fixed in the middle height of the sheet at 100 mm distance in both sides from the middle plane of the spanwise. The temperature of the unheated parts of the wall is measured by another twelve thermocouples from the same kind. The distance between each two thermocouples is 5 mm. The temperature of each of the unheated right vertical and horizontal walls is measured by two thermocouples from the same copper-constantan type which are used to measure the heating elements. The total number of thermocouples which are used to measure the temperature of the heated and unheated surfaces of the enclosure was thirty-two thermocouples. These thermocouples were connected to a Yokogawa 24-points digital temperature recorder with scale division of 0.1C through a selection connecting switch

During the course of the experimental work nearly three hours were needed to reach the steady state condition. This condition was recognized when the temperature reading did not change within a time period of about 15 minutes

The local and average heat transfer coefficients along the surface of the heater are presented by means of a local and average Nusselt numbers according to their definition for isoflux heating as:

$$Nu = h.L / k = (q.L / k) / (T - T_w)$$

$$\overline{Nu} = (q\,L / k)/(\overline{T} - T_w)]$$

where T_w and $\overline{T}$ are the mean cooling wall temperature and the average heater surface temperature, respectively.

The other dimensionless parameters which are used in the presentation of the results in this study were the Prandtl number Pr and the Rayleigh number Ra:

$$Pr = \nu/\alpha \quad , \qquad \text{and} \quad Ra = g\beta\, L^4\, q/(k\alpha\nu)$$

Where k, ν, α, and β are the fluid thermal conductivity, kinematic viscosity, thermal diffusivity, and thermal expansion coefficient, respectively.

RESULTS AND DISCUSSION

Two heating elements were placed in the cavity on the vertical wall and studied experimentally for an enclosure aspect ratio A=2, Prandtl number Pr=0.7 and different values of Rayleigh number Ra = 0.433×10^5, 1.73×10^5, 2.6×10^5, 3.68×10^5, 4.9×10^5 and 6.29×10^5. Three cases were considered, with different heating element locations. For case (1) the heater locations are $B_1/H=0.2$ and $B_2/H=0.5$, for case (2) $B_1/H=0.2$ and $B_2/H= 0.8$ and for case (3) $B_1/H=0.5$ and $B_2/H=0.8$. The effect of the heating element positions, wake distance between the lower and upper heating elements, and Rayleigh number on the local and maximum surface temperatures and the local and average Nusselt number were presented.

Effect of the lower heating element position

Figures 3 and 4 present the effect of the lower heater position on both the local surface temperature and the local Nusselt number for Ra= 0.433×10^5, 1.73×10^5, 3.68×10^5 and 6.29×10^5. In these Figures, case (1) and case (3) are considered. They have equal wake distances between the lower and upper heaters and different lower heater position of $B_1/H=0.2$ and 0.5 respectively. The local surface temperatures of both heaters of case (1) are smaller than those of case (3). While, the local Nusselt number of both heaters for case (1) are higher than those of case (3).

Figures 5 and 6 show the behavior of both the maximum temperatures and the average Nusselt number for both the lower and upper heaters, with the variation of Rayleigh number for the three considered cases. Considering cases (1) and (3), Figures 5 and 6 show that for constant Rayleigh number, the maximum temperature of the lower and upper heaters for case (1) are smaller than those of case (3). While the average Nusselt number of the lower and upper heaters for case (1) are larger than those of case (3).

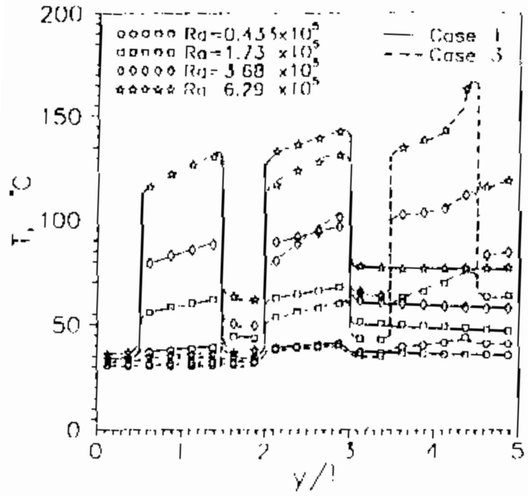

Fig. 3 Relation between the surface temperature versus vertical distance for cases (1) and (3)

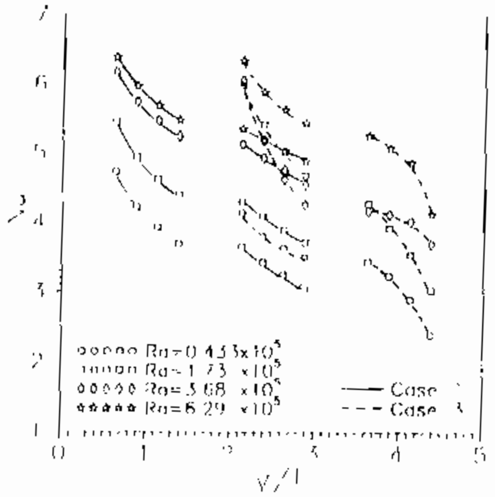

Fig. 4 Relation between the local Nusselt number versus vertical distance for cases (1) and (3)

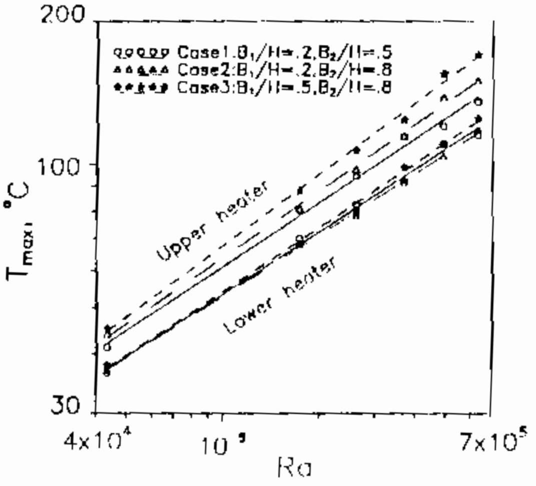

Fig 5 Relation between the maximum temperature versus Rayleigh number for cases (1), (2) and (3)

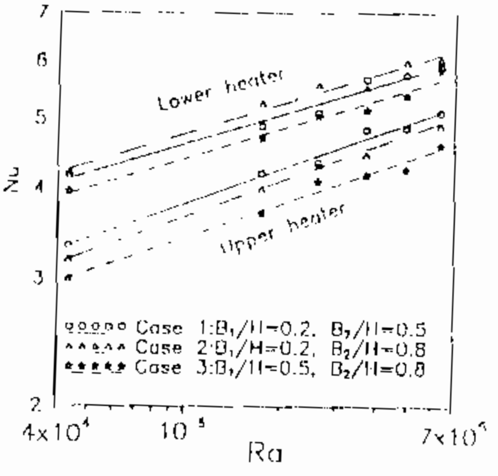

Fig 6 Relation between the average Nusselt number versus Rayleigh number for cases (1), (2) and (3)

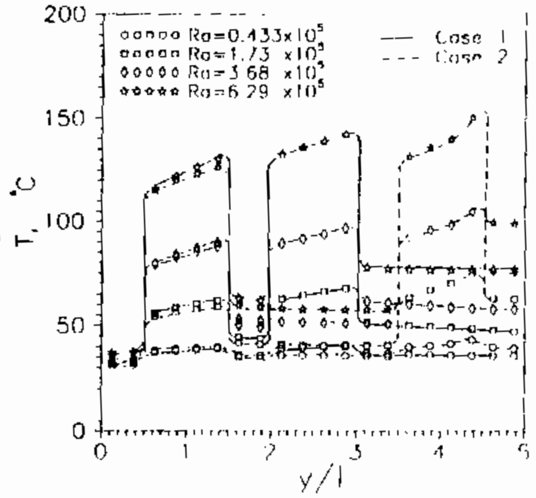

Fig. 7 Relation between the surface temperature versus vertical distance for cases (1) and (2)

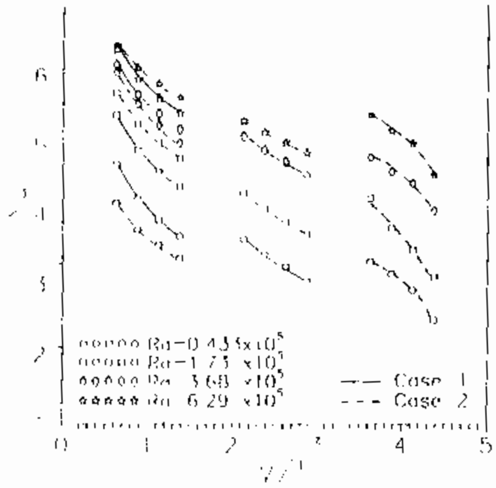

Fig. 8 Relation between the local Nusselt number versus vertical distance for cases (1) and (2)

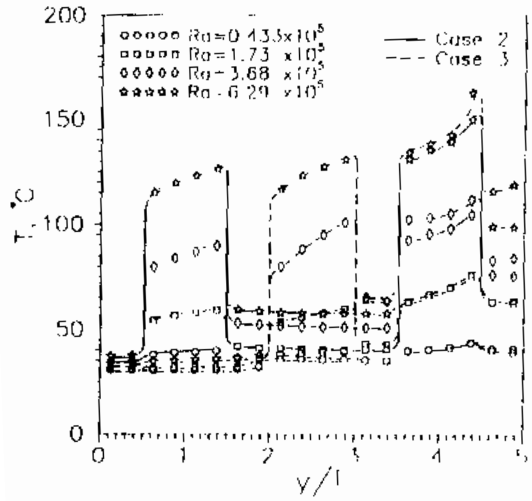

Fig. 9 Relation between the surface temperature versus vertical distance for cases (2) and (3)

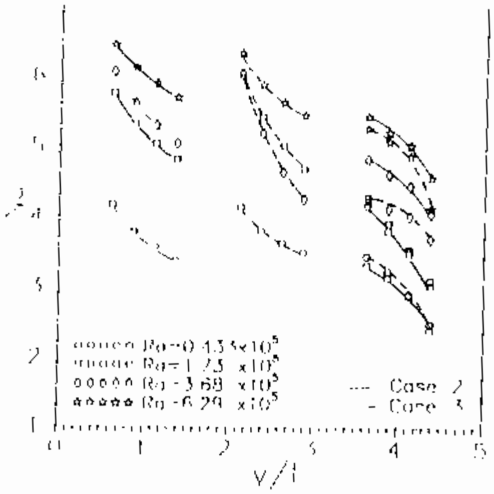

Fig. 10 Relation between the local Nusselt number versus vertical distance for cases (2) and (3)

For constant wake distance and Rayleigh number, as the lower heater is moved nearer to the lower horizontal wall, the local and maximum temperatures for the lower and upper heaters decreases and the local and average Nusselt number for the lower and upper heaters increases which is the required case

Effect of upper heating element position

Figures 3 and 4 show the effect of the upper heater position on both the local surface temperature and Nusselt number for the upper and lower heater . The wake distance between the heaters are fixed and the upper heater position varies from $B_2/H=0.5$ to 0.8 for cases (1) and (3) respectively. Figures 3-6 show that for constant wake distance and Rayliegh number, with the decrease of B_2/H, i.e. as the upper heater becomes more far from the upper horizontal wall the heater surface local and maximum temperatures decrease while the local and average Nusselt number increases which is the required case.

Effect of wake distance

Cases (1) and (2) were considered. The lower heater has fixed position with $B_1/H=0.2$, while the wake distance between the two heaters increases to make B_2/H increases from 0.5 to 0.8. Figures 7 and 8 represent the effect of wake distance on the local surface temperature and Nusselt number for $Ra= 0.433\times10^5$, 1.73×10^5, 3.68×10^5 and 6.29×10^5. Figures 7-8 show that for constant Rayleigh number and fixed lower heater position, as the wake distance increases the upper heater becomes nearer to the upper horizontal wall, the local temperature of the upper heater increases which increases the temperature of the flow reaching the vertical cold wall, the flow leaving the cold wall and the flow reaching the leading edge of the lower heater Therefore, the local temperature of lower heater increases marginally, which in turns, decreases the local Nusselt number. These results are supported by the results of Figs 5 and 6. In which the maximum surface temperature of the lower heater increases and the average Nusselt number decreases with the increase of the wake distance.

To show the effect of the wake distance on the upper heater, cases (2) and (3) were considered. The position of the upper heater is fixed with $B_2/H=0.8$ and the wake distance between the heaters decreases to make B_1/H increases from = 0.2 to 0.5 respectively. It is shown from Figs 9 and 10 that with the increase of the wake distance, i.e as the lower heater is moved more far from the upper heater, the local surface temperature of the upper heater decreases and the local Nusselt number increases. Figures 5 and 6 show also that for cases (2) and (3), with the increase of the wake distance between the upper and lower heater the maximum surface temperature of the upper heater decreases and the Nusselt number increases These results supports that the position of the upper heater must be as far as possible from the upper horizontal wall and as far as possible from the lower heater. A compromise must be done to make the best position for the upper heater to be in the middle of the distance between the lower heater and the upper horizontal wall.

Effect of Rayleigh number

Figures 3, 4, and 7 to 10 present the effect of Rayleigh number variation on both the local surface temperature and Nusselt number of the upper and lower heater.

With the increase of Rayleigh number both the local surface temperature and Nusselt number increase for the two heaters. As shown in Figs 5 and 6, the maximum surface temperature and the average Nusselt number for both heaters increases as the Rayleigh number increases. The maximum heater surface temperature and the average Nusselt number for both the upper and lower heaters are correlated as a function of Rayleigh number by equations as follows.

$$T_{max} = c \ Ra^{d} \ , \quad \overline{Nu} = m \ Ra^{n}$$

where c, d, m, and n are constants and are expressed in the following table

	Lower heater				Upper heater			
	c	d	m	n	c	d	m	n
case (1)	0.356	0.435	0.990	0.134	0.367	0.444	0.612	0.519
case (2)	0.358	0.428	0.989	0.137	0.310	0.463	0.558	0.164
case (3)	0.324	0.445	0.923	0.136	0.224	0.496	0.578	0.155

CONCLUSIONS

The effect of the lower and upper heater positions, the wake distance between them and Rayleigh number on the heaters surface temperature and heat transfer characteristics are studied experimentally. The considered parameters are Pr=0.7, and Ra = 0.433×10^{5}, 1.73×10^{5}, 2.6×10^{5}, 3.68×10^{5}, 4.9×10^{5} and 6.29×10^{5}. The results show the following conclusions.

For constant wake distance and Ra, as the lower heater is moved nearer to the lower horizontal wall, T and T_{max} for the lower and upper heaters decrease while Nu and $\overline{Nu}$ for the lower and upper heaters increase which is the required case. As the upper heater becomes more far from the upper horizontal wall the heater surface T and T_{max} decrease while Nu and $\overline{Nu}$ increase which is the required case.

For constant Rayleigh number, as the wake distance increases with fixed lower heater position, T and T_{max} increase marginally for the lower heater, which in turns, decreases Nu and $\overline{Nu}$. For fixed upper heater position, the increase of the wake distance, decreases T and T_{max} and increases Nu for the upper heater.

To get minimum T and T_{max} and maximum Nu and $\overline{Nu}$ for the lower and upper heaters, the position of the lower heater must be nearer to the lower horizontal wall and the position of the upper heater must be as far as possible from the upper horizontal wall and as far as possible from the lower heater. A compromise position for the upper heater is in the middle of the distance between the lower heater and the upper horizontal wall.

With the increase of Rayleigh number, T, T_{max}, Nu and $\overline{Nu}$ increase for both heaters. T_{max} and $\overline{Nu}$ for both the upper and lower heaters are correlated as a function of Rayleigh number by equations as follows.

$$T_{max} = c \ . \ Ra^{d} \ , \quad \overline{Nu} = m \ Ra^{n}$$

where c, d, m, and n are constants

NOMENCLATURE

A	aspect ratio, H/W
B_1	height of the lower heater center, m
B_2	height of the upper heater center, m
g	gravitational acceleration, m/s^2
H	enclosure height, m
k	fluid thermal conductivity, W/mK
L	heater height, m
Nu	local Nusselt number along the heater surface, Eq. (9)
$\overline{Nu}$	average Nusselt number, Eq. (10)
Pr	Prandtle number, ν/α
q	heat flux at the heater surface, W/m^2
Ra	Rayleigh number, $Ra = g\,\beta\,L^4\,q/(k\alpha\nu)$
T	temperature, C
T_w	mean temperature for the cooled wall, C
W	enclosure width, m
α	fluid thermal diffusivity coefficient, m^2/s
β	fluid volumetric coefficient of thermal expansion, 1/K
ν	fluid kinematic viscosity of the fluid, m^2/s
ρ	fluid density, kg/m^3

REFERENCES

1. Jaluria, Y., "Natural Convective Cooling of Electronic Equipment," in "Natural convection, Fundamentals & Applications" Kakac, S., Aung, V., and Viskanta, R., Hemisphere Publishing corporation, pp. 961-986, 1985.
2. Incropera, F., P., "Convection Heat Transfer in Electronic Equipment Cooling," ASME J. of Heat Transfer, Vol. 110, pp. 1097-1111, 1988
3. Papanicolaou, E., and Jaluria, Y., "Mixed Convection From Simulated Electronic Components at Varying Relative Positions in a Cavity," ASME J. of Heat Transfer, Vol. 116, pp. 960-970, 1994.
4. El Kady, M. S., and Araid, F. F., "Natural Convection From a Single Surface Heater in a Vertical Rectangular Enclosure," Mansoura Engineering Journal (MEJ), Vol. 23, No. 2, pp M -M , June.1998.
5. Jaluria, Y., "Buoyancy-Induced Flow Due to Isolated Sources on a Vertical Surface," ASME J. of Heat Transfer, Vol. 104, pp. 223-227, 1982
6. Anand, N., Kim, S., and Aung, W., "Effect of Wall Conduction of Free Convection Between Asymmetrically Heated Vertical Plates: Uniform Wall Temperature," Int J Heat Mass Transfer, Vol. 33, pp. 1025-1028, 1990.
7. Kim, S., Anand, N., and Aung, W., "Effect of Wall Conduction of Free Convection Between Asymmetrically Heated Vertical Plates: Uniform Wall Heat Flux," Int. J. Heat Mass Transfer, Vol. 33, pp. 1013-1023, 1990.

8. Kim, S., Anand, N., and Fletcher, L., "Free Convection Between Series of Vertical Parallel Plates With Embedded Line Heat Sources," ASME J. of Heat Transfer, Vol. 113, pp. 108-115, 1991
9. Keyhani, M., Prasad, M., and Cox, R., "An Experimental Study of Natural Convection in a Vertical Cavity with Discrete Heat Sources," ASME J. of Heat Transfer, Vol. 110, pp. 616-624, 1988.
10. Keyhani, M., Chen, L., and Pitts, D., "Aspect Ratio Effect on Natural Convection in an Enclosure With Protruding Heat Sources," ASME J. of Heat Transfer, Vol. 113, pp. 883-891, 1991.
11. Carmona, R., and Keyhani, M., "The Cavity Width Effect on Immersion Cooling of Discrete Flush-Heaters on One Vertical Wall of an Enclosure Cooled From the Top," ASME Journal of Electronic Packaging, Vol. 111, pp. 268-276, 1989