A NUMERICAL STUDY OF NATURAL CONVECTIVE HEAT TRANSFER FROM A PAIR OF ADJACENT HORIZONTAL ISOTHERMAL SQUARE ELEMENTS EMBEDDED IN AN ADIABATIC SURFACE – EFFECT OF ELEMENT SPACING ON HEAT TRANSFER RATE

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ABSTRACT

In natural convective heat transfer from two horizontal heated surfaces, if the surfaces are mounted relatively close together the flows over the adjacent surfaces can interact and this interaction can affect the heat transfer from the surfaces. In the present numerical study this has been investigated by considering two adjacent square horizontal isothermal surfaces or elements. The purpose of the study was to determine whether the flow interaction does have a significant effect on the heat transfer rate and, if it does, at what distance between the element does the flow interaction start to affect the heat transfer rate to a significant extent. Two adjacent square plane horizontal isothermal elements of the same size embedded in a horizontal plane adiabatic surface have been considered. It is assumed that the surfaces of the square heated elements are in the same plane as the surrounding adiabatic surface. Attention has been restricted to the case where the heated elements are facing upward. The heat transfer from the elements has been assumed to be to air because of the applications being considered. Steady flow has been assumed. Constant fluid properties have been assumed except for the density change with temperature which gives rise to the buoyancy forces. This was treated using the Boussinesq approach. For the range of conditions considered here laminar, transitional, and turbulent flows over the elements can occur. The solution has been obtained by using the commercial CFD solver ANSYS FLUENT© to numerically solve the governing equations subject to the boundary conditions. The *k*-epsilon turbulence model has been used. The sides of the square heated elements are assumed to be parallel to each other. The Nusselt number based on the side length of the square elements will depend on Rayleigh number, on the dimensionless distance between the adjacent sides of the elements and on the Prandtl number. Since heat transfer to air is considered results have been obtained only for a Prandtl number of 0.74. The effect on the variation of the Nusselt number with Rayleigh number of the dimensionless distance between the sides of the element has been explored and the conditions under which the effect of the interaction between the flows over the adjacent elements can be ignored have been examined.

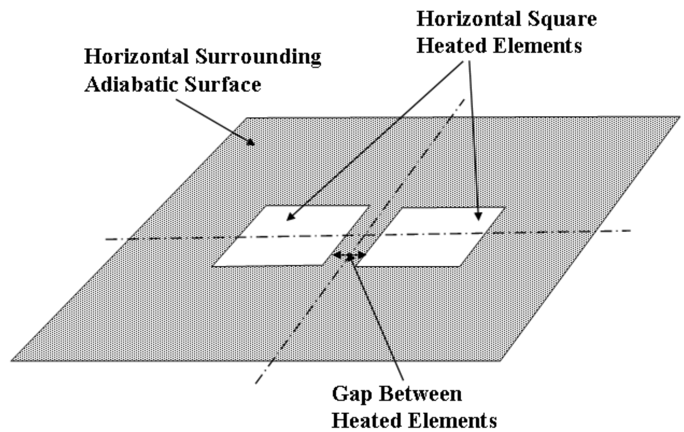

Figure 1 Flow situation considered

INTRODUCTION

In some situations that can arise in the natural convective air-cooling of non-computer electronic devices in various industrial applications the components are mounted horizontally and relatively close together. In such cases the interaction of the flows over the adjacent components can affect the heat transfer rate from the components. The purpose of the present numerical study was to investigate the distance between adjacent surfaces of the components that is likely to affect the heat transfer rate and the nature of this effect. A simplified model of the real situation has been considered here. The components are represented by two adjacent square plane horizontal isothermal elements of the same size embedded in a horizontal plane adiabatic surface as shown in Fig. 1, the arrangement of the elements being as shown in Fig. 2. The surfaces of the heated elements are assumed to be in the same plane as the surrounding adiabatic surface. Attention has been restricted to the case where the heated elements are facing upward. The heat transfer from the element has been assumed to be to air because of the applications being considered. For the range of conditions considered here laminar, transitional, and turbulent flows can occur over the elements.

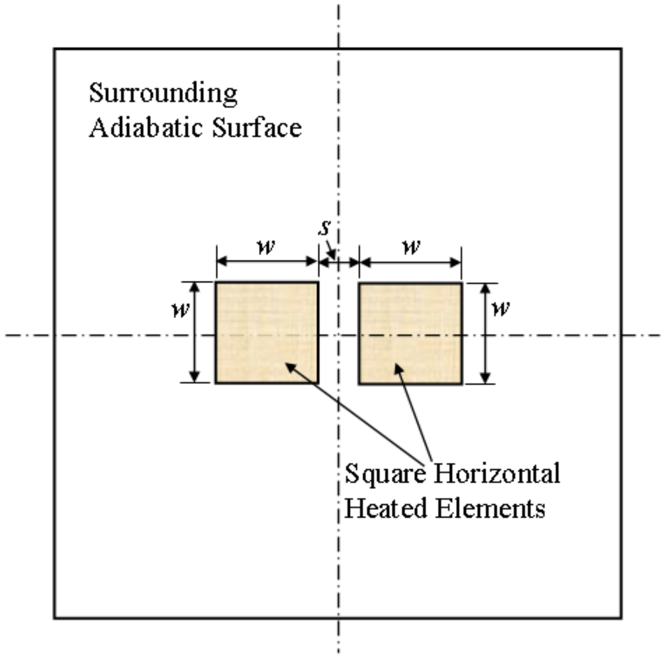

Figure 2 Variables used in defining flow situation

Natural convective heat transfer from single upward-facing heated horizontal surfaces has been extensively studied, e.g., see [1-14]. Most of these studies, however, have considered only situations that involve laminar flow. The present study differs from these available studies by considering multiple adjacent elements and by obtaining results for a wide range of Rayleigh numbers that include conditions in which laminar, transitional, and turbulent flow occur. This study is one of a series of investigations of natural convective heat transfer rates from various horizontal and inclined heated element arrangements, e.g. see [15-19].

NOMENCLATURE

A	[m^2]	Surface area of heated element
g	[m/s^2]	Gravitational acceleration
k	[W/mK]	Thermal conductivity
Nu	[-]	Mean Nusselt number based on w
Nu_l	[-]	Local Nusselt number based on w
Pr	[-]	Prandtl number
$\overline{Q'}$	[W]	Mean heat transfer rate from element
q'	[W/m^2]	Local heat transfer rate per unit area
Ra	[m]	Rayleigh number based on w
s	[m]	Gap between elements
S	[-]	Dimensionless gap between elements, s/w
T_f	[K]	Undisturbed fluid temperature
T_w	[K]	Heated element wall temperature
w	[m]	Side length of square heated element

Greek Symbols

α	[m^2/s]	Thermal diffusivity
β	[1/K]	Bulk coefficient of thermal expansion
ν	[m^2/s]	Kinematic viscosity

SOLUTION PROCEDURE

Steady flow has been assumed. Fluid properties have been assumed to be constant except for the density change with temperature giving rise to the buoyancy forces. The Boussinesq approach was employed in order to deal with this.

The solution has been obtained by using the commercial CFD solver ANSYS FLUENT© to numerically solve the governing equations subject to the boundary conditions. The *k*-epsilon turbulence model has been used here with full account taken of buoyancy force effects. A number of studies, e.g., [20-24], have shown that this method gives quite good predictions of the conditions under which turbulence develops and of the heat transfer rate with transitional and turbulent flows.

The sides of the heated elements are assumed parallel to each other and the flow is assumed symmetrical about the line mid-way between the two adjacent sides of the elements and about the centre-lines of the elements shown in Fig. 2.

The heat transfer rate from the heated element expressed in terms of a Nusselt number is dependent on the Rayleigh number, on the size of the gap between the two elements relative to the overall size of the element, and on the Prandtl number. Results for a Prandtl number of 0.74, i.e., effectively the value for air, have been obtained. Rayleigh numbers between approximately 10^5 and 10^{13} have been considered.

Extensive grid independence and convergence-criteria independence testing was undertaken for this study. Results of this testing indicated that with the grids employed in obtaining the present results the derived heat transfer rates are grid- and convergence criteria independent to within about one per cent.

RESULTS

The solution has the following parameters:

- Ra, the Rayleigh number based on the side length, w, of the square elements (see Fig. 2) and the difference between the temperature of the surface of the heated elements, T_w, and the temperature of the undisturbed fluid well away from the system, T_f, i.e.:

$$Ra = \frac{\beta g w^3 (T_w - T_f)}{\nu \alpha} \tag{1}$$

- The dimensionless distance between the adjacent sides of the two square elements (see Fig. 2), i.e. on $S = s / w$
- The Prandtl number, Pr.

Results have only been obtained for a Prandtl number of 0.74, i.e., effectively the value for air due to the applications that motivated this study. Results have been obtained for S values between 0 and 0.7. The case of $S = 0$ corresponds to the case where the elements are in contact with each other and effectively form a single rectangular element. For comparison purposes results have also been obtained for the case of an isolated square element, i.e., for the case where the two square elements are so far apart that there is no interaction between the flows over the two elements. Rayleigh numbers of between approximately 10^4 and 10^{12} have been considered.

The mean heat transfer rate from the heated elements, $\overline{Q}'$, has been expressed in terms of a mean Nusselt number that is based on the side length, w, of the square heated element and on the difference between the temperature of the surface of the heated element, T_w, and the temperature of the undisturbed fluid well away from the system, T_f, i.e.:

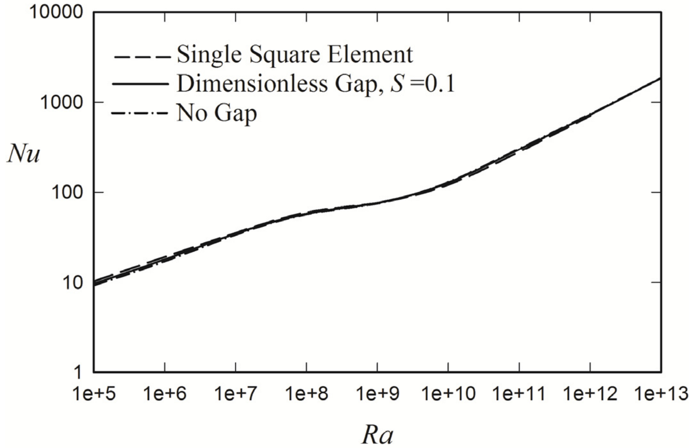

Figure 3 Variation of mean Nusselt number with Rayleigh number for a dimensionless element spacing of 0.1. For comparison the variations for the cases where there is no gap between the elements and where the elements are far apart are also shown.

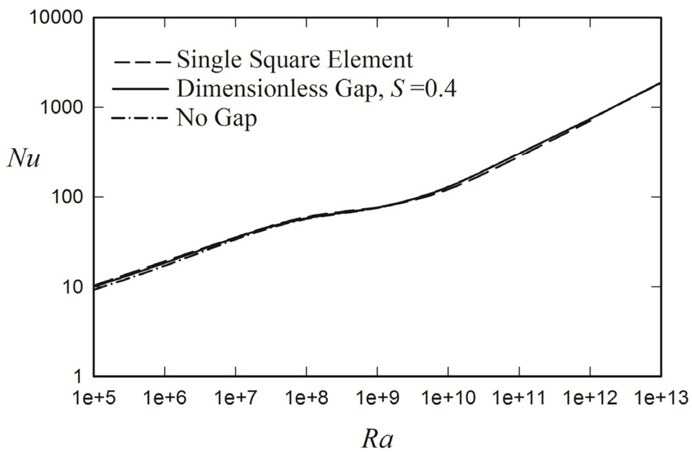

Figure 4 Variation of mean Nusselt number with Rayleigh number for a dimensionless element spacing of 0.4. For comparison the variations for the cases where there is no gap between the elements and where the elements are far apart are also shown.

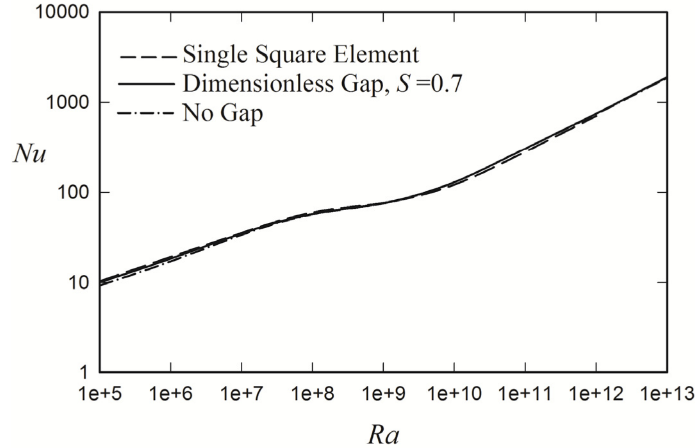

Figure 5 Variation of mean Nusselt number with Rayleigh number for a dimensionless element spacing of 0.7. For comparison the variations for the cases where there is no gap between the elements and where the elements are far apart are also shown.

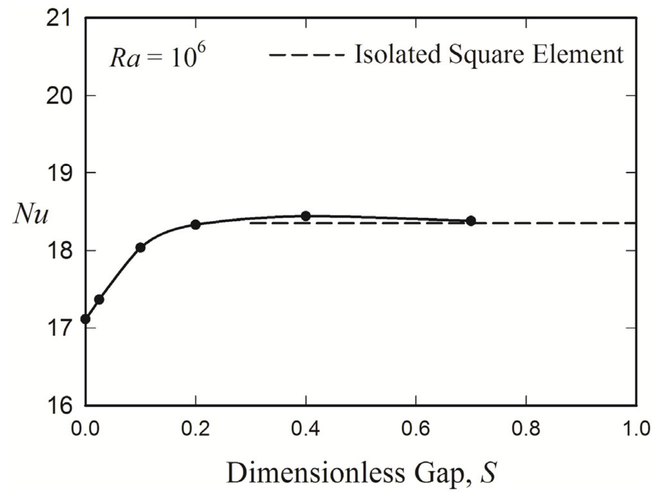

Figure 6 Variation of mean Nusselt number with dimensionless element spacing for a Rayleigh number of 10^6.

$$Nu = \frac{\bar{Q}'w}{kA\,(T_w - T_f)} \tag{2}$$

where A is the surface area of the heated surface, i.e., w^2 . Since this study considers a fixed value of *Pr, Nu* is a function of *Ra* and the dimensionless spacing between the elements, *S*.

Typical variations of the mean Nusselt number with Rayleigh number for three different values of the dimensionless spacing between the elements are shown in Figs. 3, 4, and 5. Also shown in these figures are the variations for the limiting cases where the elements are in contact, i.e., $S = 0$, and where S is very large and where there is, therefore, no interaction of the flows over the elements.

It will be seen from the results given in Figs. 3, 4, and 5 that the effect of the flow interaction on the heat transfer rate for the two square element situations being considered in the present study is relatively small. This is further illustrated by the results given in Figs. 6, 7, and 8 which show the variations of the Nusselt number with the dimensionless element spacing for three Rayleigh number values. It will be observed that the form of the variation of the Nusselt number with the dimensionless element spacing is strongly dependent on the Rayleigh number. From the results given in Figs. 6, 7, and 8 it will be seen that the largest percentage difference between the highest and lowest Nusselt number values for a given Rayleigh number is less than 10%.

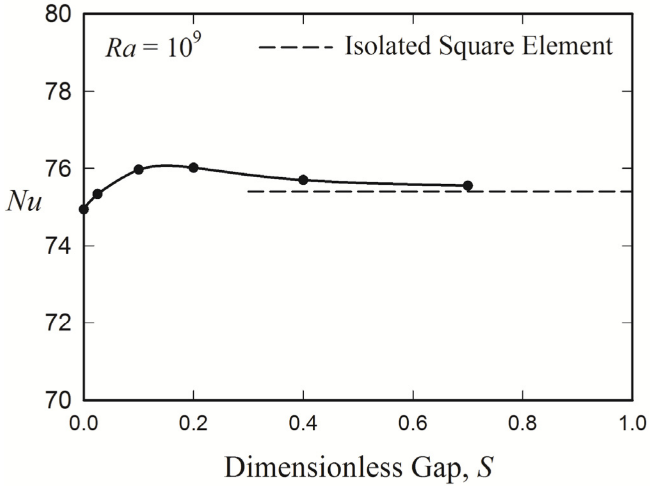

Figure 7 Variation of mean Nusselt number with dimensionless element spacing for a Rayleigh number of 10^9.

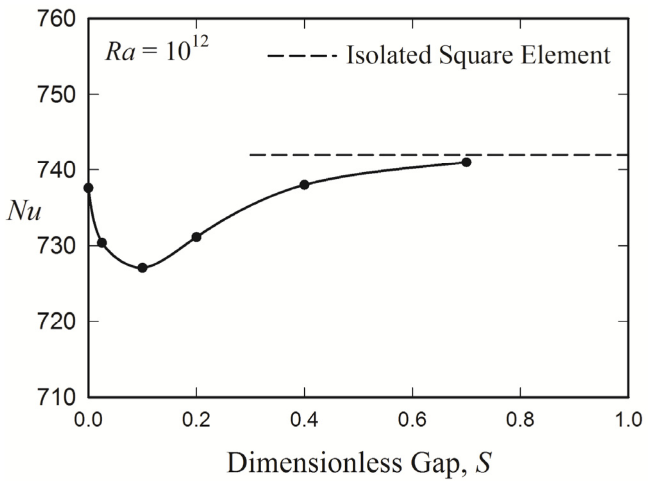

Figure 8 Variation of mean Nusselt number with dimensionless element spacing for a Rayleigh number of 10^{12}.

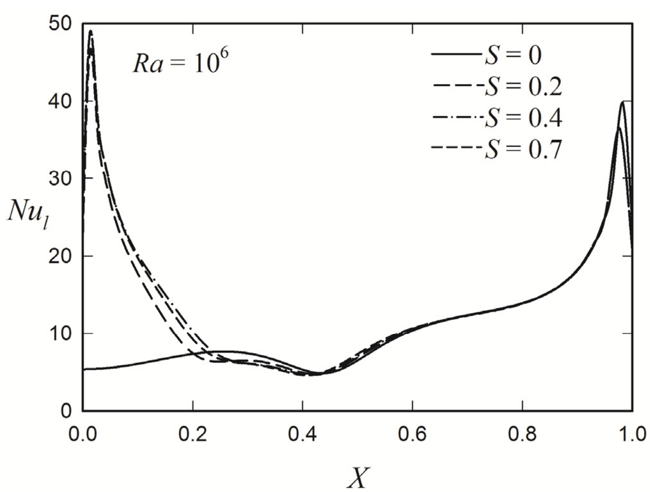

Figure 9 Variation of local Nusselt number with dimensionless distance along element centre-line for a Rayleigh number of 10^6 for various values of the dimensionless element spacing, *S*.

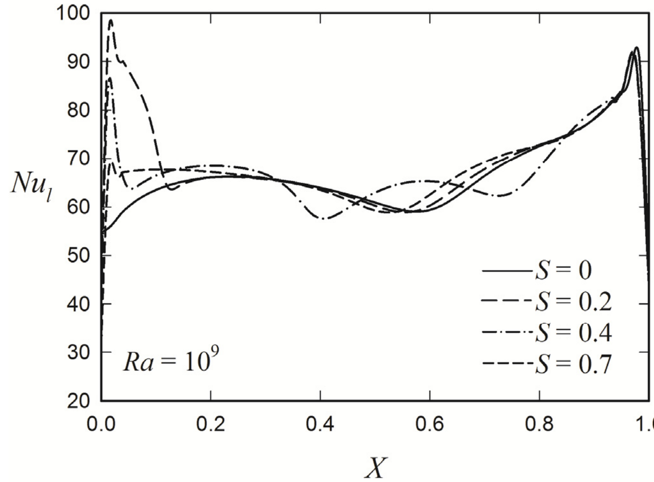

Figure 10 Variation of local Nusselt number with dimensionless distance along element centre-line for a Rayleigh number of 10^9 for various values of the dimensionless element spacing, *S*.

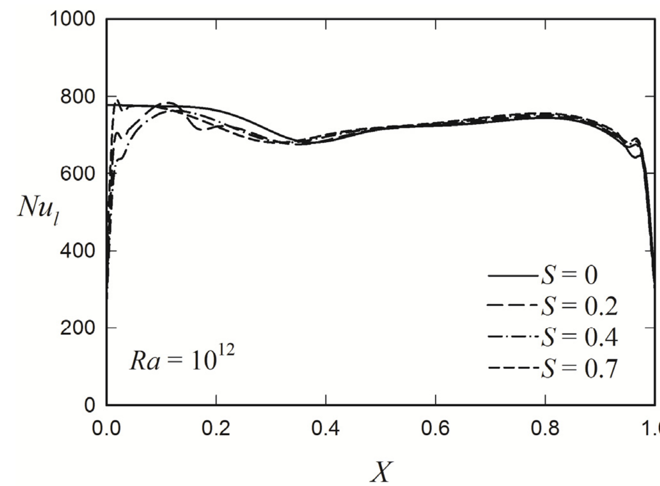

Figure 11 Variation of local Nusselt number with dimensionless distance along element centre-line for a Rayleigh number of 10^{12} for various values of the dimensionless element spacing, *S*.

While the spacing between the square elements has a relatively weak effect on the mean heat transfer rate it has a more significant effect on the local heat transfer rate distribution over the surface of the elements and these changes can have effects that are important in some practical situations. This local heat transfer rate distribution is dependent on the flow patterns over the two adjacent elements. When the elements are close together the upward flows from the two plates merge into a single upward plume above the gap between the elements while when the elements are far apart there is little interaction between the flows above each element and the flow consists mainly of an upward plume above the centre of each element. The actual flow pattern is also dependent on the Rayleigh number. These changes in the flow pattern over the elements are related to the local heat transfer rate distribution over the elements. Typical variations of the local Nusselt number based on w, i.e.:

$$Nu_l = \frac{q'w}{k(T_w - T_f)} \tag{3}$$

along the centre-line of an element for different dimensionless element spacing and various Rayleigh numbers are shown in Figs. 9, 10, and 11. The form of the local Nusselt number variation will be seen to be strongly dependent on both the Rayleigh number and the dimensionless element spacing, the local Nusselt numbers changing significantly near the element edge that is adjacent to the edge of the other element.

CONCLUSIONS

The results of the present study indicate that:

1. The dimensionless spacing between the two heated elements has a relatively small effect on the mean element Nusselt numbers, the maximum change in the mean Nusselt numbers with changes in the dimensionless element spacing at a given Rayleigh number being less than 10%.
2. The form of the variation of the mean element Nusselt number with dimensionless element spacing is strongly dependent on the Rayleigh number.
3. The dimensionless spacing between the square elements has a significant effect on the local heat transfer rate distribution over the surface of the elements, the nature of this effect being relatively strongly dependent on the Rayleigh number.

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