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EXPERIMENTAL STUDY OF GEOMETRIC EFFECTS ON FORCED AIR-COOLING OF REGULAR IN-LINE ARRAY OF ELECTRONIC COMPONENTS

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ABSTRACT

Systematic experiments were performed to investigate effects of most influential geometric parameters (channel height, components' height, inter-components' spacing, and heated component row number), on the heat transfer coefficient of a single heated rectangular component placed in a quasi-regular in-line array of electronic components. The air approach mean velocity ranged from 1.7 to 10.5 m/s, corresponding to component length Reynolds numbers from 2800 to 17200. Both sparse and dense array of simulated components were used. Channel height ranged from 1.5 to 3.0 times of component height, while inter-components spacing varied from 0.43 to 1.0 times of component length. Both cubical and flatpack components with different heights were used and the heat transfer coefficient varied from 32 to 224 W/m^2K. The experimental results indicate that the heat transfer coefficient was mainly affected by the geometric parameters and approach Reynolds number. Based on the obtained results, a general empirical heat transfer correlation was developed.

INTRODUCTION

While forced air cooling continues to be adequate for many relatively high speed digital electronics, techniques for prediction of the component operating temperature are actively being investigated for future generations of electronic equipment such as high performance computers. Sufficient knowledge of the convective heat flux distribution over the exposed surfaces of an electronic printing wiring board is required as a guide to accurately predict the operating temperature of the heated component. The convective heat flux, in turn, is mainly affected by the array geometry and the air flow rate.

Several experimental investigations related to the effects of geometry on forced air-cooling of electronic equipment have been conducted over the past decade. An extensive review of this literature have been compiled by Arabzadeh (1993). For forced convective heat transfer of a heated component in a regular in-line array of rectangular components, the relevant geometric parameters are H/t, S/L, t/L, and heated component row number (r). None of the reported correlations included the effects of all of these four geometric key parameters. Since the arrays tested were different in geometry as well as size, the urge to combine the reported correlations and deduce a single correlation which covers the effects of all these four parameters is also impossible. Therefore, the reported correlations work only for the experimenter's experimental setup, or with limitations for setups that are somewhat similar.

Arvizu (1981), Arvizu and Moffat (1982), and Anderson and Moffat (1990), reported a superposition method to calculate the temperature rise of a rectangular component due to the effects of thermal wakes of its upstream heated components. It should be noted that during our data collection, the temperature rise of components downstream of the heated component were also measured and recorded in order to compare with their data for the superposition method. This comparison revealed and verified that their suggested superposition correlation is accurate enough (within our experimental uncertainty) for only predication of the temperature rise of any component due to the thermal wakes of its upstream heated components, in a regular in-line array of rectangular components. However, another comprehensive correlation is needed to predict the self-heating temperature rise of the component due to its internal heating, which is a function of all relevant geometric parameters. This can be the proposed correlation of this study. Summation of these two temperature rises gives the total temperature rise of the component. Their suggested correlation for prediction of self-heating temperature rise is only limited to their tested array, since the effects of important parameters such as components' height (t), and heated component row number (r) were not included.

Sparrow et al. (1982) reported heat transfer data for a regular in-line array of "flatpacks". They used the naphthalene sublimation technique, with the heat transfer Nusselt number deduced from the measured Sherwood number by invoking the mass transfer analogy. They reported row-independent (fully developed) heat transfer coefficients for fifth and all subsequent rows. None of the important geometric parameters appear in their suggested correlation. Furthermore, since they used the analogy between mass and heat transfer, the accuracy of their results should be verified by actual temperature measurements.

Wirtz and Dykshoorn (1984) reported heat transfer studies for an in-line array of "flatpacks". The components were arranged in a square array in both the streamwise and the spanwise directions. Using conventional temperature measurement techniques, the experimental results showed that the thermal wakes from heat dissipating components have significant influence on heat transfer of downstream components. However, the effects of the hydrodynamic wake and components' height were not investigated separately. It was found by Lehmann and Wirtz (1985) that the convection from the component surface increased when component spacing was increased.

Moffat et al. (1985) with an experimental setup similar to that used by Arvizu and Moffat (1982), presented heat transfer coefficients and thermal wake functions for in-line arrays of cubical components mounted on one wall of a parallel planar channel for several combinations of channel height (plate spacing) and approach velocity. It was found that the change in channel height can result in different flow patterns and hence, large variations in the heat transfer coefficient. However, they did not investigate effect of components' height and heated component row number, on convective heat transfer coefficient of the heated component.

Tai and Lucas (1985) tested components of three different sizes in a package, as did Biber and Sammakia (1986). The package used in both studies differed from previous work in its lack of entrance and exit channel. They fitted their data to an equation different than a simple exponential . Biber and Sammakia also employed three modified exponential correlations depending on the three different arrays. However, they did not investigate the effects of each geometric parameter individually.

Chang et al. (1987) investigated the influence of the hydrodynamic wake from one unheated component on the heat transfer of a downstream heated component. Only two components were used in their experiments. It was found that the hydrodynamic wake from the upstream component can cause a large variation in heat transfer on the heated component, depending on the spacing between the two components. They also showed that the average heat transfer coefficient from the heated component can be expressed by the Colburn j-factor, which is a function of the Reynolds number and the ratio of channel height to component height. Since their test section was composed of only two rectangular components, effects of components' spanwise spacing, as well as components' height were not included in their offered correlation. Their results can not be used for fully populated arrays.

Copeland (1992) performed a series of experiments to study the effects of channel height, components' spacing, and heated component row number on forced convection of a regular in-line array of "flatpacks". Their suggested correlations are limited to their tested arrays, since effects of components' height were not taken into account.

More recently, Wirtz and Mathur (1993) studied the distribution of convective heat transfer over the five exposed faces of an electronic package. They used very "flatpack" components ($t/L = 0.17$) and a dense array ($S/L = 0.5$). They reported that the average heat transfer coefficient for the top surface was found to be nearly equal to the overall average heat transfer coefficient for the element. They concentrated on one component's surface heat transfer coefficient rather than the general trend of component's heat transfer coefficient variation caused by the changes of geometric parameters.

In spite of having expanded heat transfer database in the area of forced convective electronic cooling, there is still no single general correlation available that can predict the heat transfer behavior of any single heated rectangular component placed in any in-line array of unheated similar rectangular components having arbitrary geometries. Lack of this generality in the reported correlations established the need for the present research. The objective of this study was to conduct systematic experiments with different ranges of the above mentioned four geometric key parameters, in order to present a single general correlation which covers the effects of all these parameters. Furthermore, the results of this study can provide an accurate and reliable experimental heat transfer database which can be used as an input for the Computational Fluid Dynamic (CFD) benchmark problems, in order to judge the adequacy of the mathematical models used.

EXPERIMENTAL SETUP AND PROCEDURES

Different parts of a versatile experimental setup were carefully designed and constructed, in order to have the ability to perform experiments for different ranges of geometric parameters, as well as accurate control and measurement of the channel average velocity, heated component temperature, and input power to the selected component(s). A schematic diagram of the apparatus used in the experiments is shown in Fig. 1. The test components were mounted along the bottom wall of a 152.4 cm long rectangular duct with a fixed 25.4 cm width and a height that was easily adjustable from 1.27 to 7.62 cm. The rectangular duct was constructed of 1.27 cm commercial grade plexiglas. Ambient air entered the rectangular duct through a 101 cm long wooden contraction which was designed to provide smooth flow of entering air. The inlet to outlet ratio (contraction ratio) varied from 14.5 to 82, depending on the channel height. For any desired channel height the appropriate flow straightener at the entrance to the rectangular duct provided a uniform velocity profile. Flow straighteners (0.678 open area ratio) consisted of soda straws (0.55 cm inside diameter, 12 cm length) tightly packed between galvanized steel mesh screens (0.044 cm wire diameter, 0.32 cm mesh width). The rectangular duct consisted of three sections: a 76.2 cm entrance section, a 38.1 cm test section, and a 38.1 cm exit section. The test section was designed such that its upper wall could be removed and reset in place quickly, thus enabling rapid access to the array of components. The entire channel flow was covered with a 1.6 mm layer of epoxy resin plate mixed with fiberglass (NEMA-G-11 manufactured by Polypenco, Inc.), which is close to what is commonly used in an actual computer board.

To accommodate variation in the height of the channel, the setup was constructed such that the bottom part of the contraction with the entire channel floor, along with the plexiglas flap attached to the inlet part of the plenum, could all be moved together and adjusted for the desired channel height. This design would prevent flow disturbance caused by the sharp leading edges of splitters or adjustable flap used by other investigators to adjust the channel height by moving the test section floor only. Ambient air was drawn past the test components by a 2 HP variable speed blower mounted downstream of the rectangular duct. The duct emptied to an acoustically absorbent plenum that served to isolate the test flow from the blower noise. The air velocity was measured and adjusted (via a computer controlled damper/stepper motor arrangement) in the round duct between the plenum and the blower by an MKS model 223BD differential pressure transducer and a pitot static tube. The blower was mounted within an insulated box and exhausted outside the building, allowing flow visualization experiments with smoke to be performed (Wang and Ghajar, 1991).

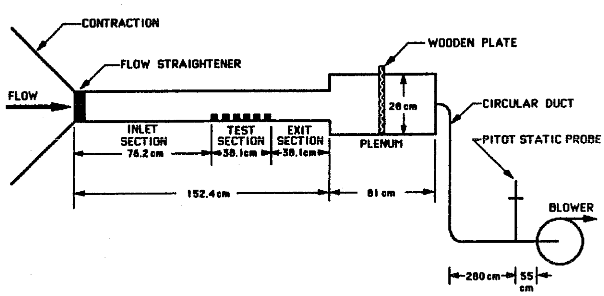

Figure 1. Schematic of Experimental Setup.

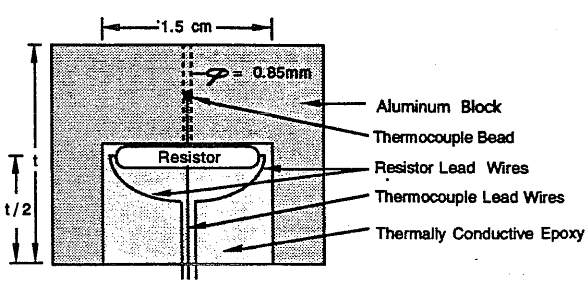

Figure 2. Detail of a Heated Component.

Electronic chips were modeled by highly polished aluminum blocks (2.54 cm x 2.54 cm x t) with a thermal conductivity of 216.3 W/m·K. The heights of components (t) varied from 1.27 to 2.54 cm. Each component was hollowed out from the back side and a 475 ohm ceramic resistor was embedded exactly at its center (see Fig. 2). To measure surface temperature of each component, a T-type calibrated copper/constantan thermocouple was bonded exactly at a position half-way between the top surface and the embedded resistor. The rest of cavity was filled with Omegatherm 201 thermally conductive epoxy. Thermocouple wires and electric leads of the resistor were routed through a threaded, hollowed nylon rod, which was used to hold the component to the test section floor with a nylon fastener nut.

Figure 3 shows a typical in-line array of the components in the test section. The idea underlying the use of half components on both ends was to more closely model an infinitely wide array. The range of dimensionless ratios defining the array were:

$t/L = 0.5$ to 1.0 $S/L = 0.43$ to 1.0 $H/t = 0.5$ to 2.0

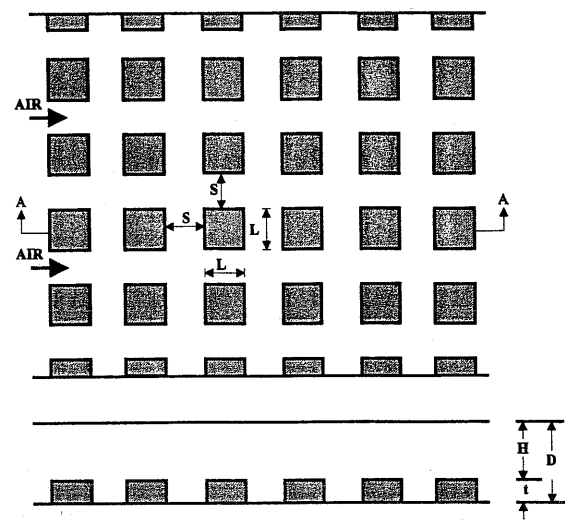

Figure 3. Typical Regular In-Line Arrangement of the Components in the Test Section.

The power delivered to each component was computer controlled. The present power controller system can handle up to forty-eight components and the power delivered to each component can be individually varied from 0 to 5 W in increments of 0.1 W. The power controller unit has a microcomputer which controls the switching circuitry and communicates via a serial port to a personal computer. Calibration tests for this unit showed that the power controller unit works with an accuracy of ±1% of the delivered power.

To relate velocity measurements in the duct to the inlet mean velocity of the air at the entrance to the test section, extensive measurements at both locations were conducted. These measurements consisted of traversing the circular duct and the rectangular channel by a pitot static tube and then numerically integrating the local velocities to obtain the mean velocity at each location. These measurements were conducted for several air velocities ranging from 1.5 to 12 m/s. From these measurements and the application of conservation of mass between the two locations, correlations which relate velocity measurements at the center of the duct to the mean air velocity at the inlet to the test section were developed. The mean air velocity at the inlet to the test section calculated from the developed relationships were compared with actual measurements for several different velocities. The calculated values in all instances were within ±2% of the measurements. Estimates of uncertainty (Kline and McClintock, 1953) show the uncertainty in the mean channel inlet air velocity varied from ±0.65%, at high velocities, to ±6.7%, at low velocities. More detail information about velocity profile, flow uniformity, and different parts of the experimental setup can be found in Arabzadeh (1993).

In the experimental procedure, the entire channel was first sealed, then the selected component was heated at the desired input power (Q_t) and the blower was turned on and adjusted through a personal computer in order to generate the required approach mean velocity. The temperature of the heated component was monitored every five minutes with a programmable forty channel datalogger interfaced with the computer, until the steady-state condition was reached when temperature variation was within ±0.5°C. It generally took 40 to 90 minutes to reach steady-state depending on approach mean velocity used. The steady-state temperature of the heated component was stored in the hard disc, then conduction and radiation losses and finally the heat transfer coefficient of the heated component were calculated. The convective heat transfer rate for each component (Q_c) was its input power (Q_t), less losses due to the conduction through the channel floor (Q_k) and thermal radiation to the surrounding (Q_r):

$$Q_c = Q_t - Q_k - Q_r \qquad (1)$$

In order to take the more accurate value of conduction loss into account, separate experiments with different board material having different thermal conductivities, were performed on the same experimental setup (see Arabzadeh, 1993; and Arabzadeh et al., 1993). These experiments demonstrated that a board composed of plexiglas and fiberglass can be considered as a non-conductive board, and one-dimensional conduction calculation is sufficient. Therefore, conduction loss was calculated by:

$$Q_k = (T_c - T_\infty)/R_W \qquad (2)$$

where T_c was the component temperature, T_∞ was the approach air temperature measured upstream of the test section, and R_W was the thermal resistance of test section floor. The net rate of thermal radiation heat exchange between the surfaces and the surroundings (Q_r) was calculated from Stefan-Boltzmann law, which in general was less than 1% of the total power supplied. Thus, the component convective heat transfer coefficient (h) was found from:

$$h = Q_c/A_c(T_c - T_\infty) \qquad (3)$$

where A_c was the heated component surface area exposed to convection.

RESULTS AND DISCUSSION

Table 1 summarizes ranges of experimental parameters that were used to investigate the influence of varying effective geometric parameters and approach Reynolds number. A single component at the center column was heated for each set of collected heat transfer data. Fixed input power of 4 watts was chosen for all of the experiments. However, preliminary experiments with different input powers were performed, in order to show that the heat transfer coefficient of the heated component was independent of the input power (Arabzadeh, 1993). For each set of experiments, the procedures outlined earlier were carefully followed. The approach flow Reynolds number, based on component length (L) and channel average velocity was $Re_L = \overline{V}_{ch}/\upsilon$; where υ is the kinematic viscosity of the air at T_∞ and $\overline{V}_{ch}$ is the approach channel mean velocity. Component length is an appropriate conventional basis for meaningful comparisons. The heat transfer results reported by some investigators, even for some set of dissimilar geometries, are possible to be compared with each other and our results by the use of component length as the characteristic length.

Estimates of uncertainty using single-sample experiments method (Kline and McClintock, 1953) showed the uncertainty in the calculated heat transfer coefficients varied from a minimum of 1.9% to a maximum of 12.5%. Repeatability was checked on the heat transfer measurements. Repeated measurements using the same heated components, the same input powers, the same instruments, the same geometric parameters and approach Reynolds numbers, on successive days showed ±1% scatter.

Heat loss due to the conduction was from a minimum of 1.9% to a maximum of 9.8%, depending strongly on the flow approach velocity and the component height, and weakly on the heated component row number and the channel height. Ranges of radiation loss was found to be from a minimum of 0.23% to a maximum of 1%, depending strongly on the approach Reynolds number, and weakly on the heated component row number and other geometric parameters.

Evidence of $h = c_g Re^n$ dependence has been observed for many years in forced convective electronic cooling. The geometric coefficient, c_g, varies with different array

Table 1. Ranges of Tested Experimental Parameters

S/L	t/L	H/t	Heated Component Row Number	Approximate Channel Average Velocity, $\overline{V}_{ch}$ (m / s)
1.0	1.0	2.0	1,2,3,4,5,6,7	2,5,10
		1.25	1,2,3,5,7,8	2,5,10
		0.5	1,2,3,4,5,8	2,5,10
	0.5	2.0	1,2,3,4,5,6	2,5,10
		1.25	1,2,3,5,7,8	2,5,10
		0.5	1,2,3,4,5	2,5,10
0.67	0.75	1.25	1,2,3,4,5,6,7	2,5,10
0.43	1.0	2.0	1,2,3,4,5,6	2,5,10
		1.25	1,2,3,4,5,6	2,5,10
		0.5	1,2,3,4,5,6	2,5,10
	0.5	2.0	1,2,3,4,5,6	2,5,10
		1.25	1,2,3,4,5,6	2,5,10
		0.5	1,2,3,4,5,6	2,5,10

geometries and is a function of r, H/t, t/L, and S/L. Figure 4 shows the variation of heat transfer coefficient as a function of Reynolds number for two different row numbers and three different channel heights. All other possible different geometric parameter cases also show the same trend for the variation of the heat transfer coefficient, which conveys the fact that there is a regular increase (a slope very near to a straight line in log-log coordinates) in the value of heat transfer coefficient, corresponding to a regular increase in the value of Reynolds number. This fact is evident since higher Reynolds number means more air movement around the heated component which causes more heat dissipation.

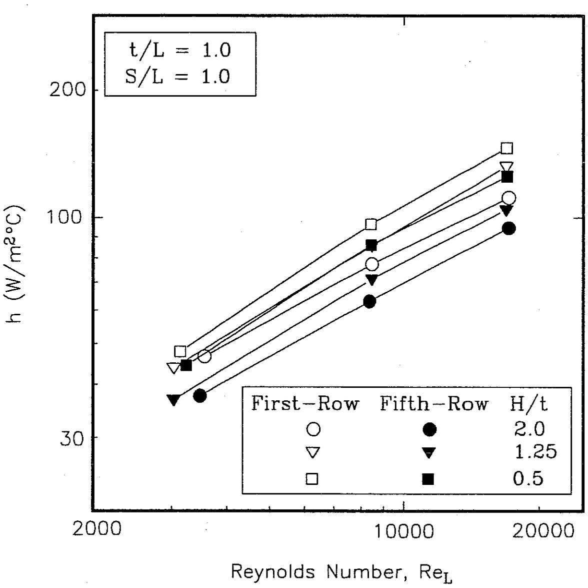

Figure 4. Heat Transfer Coefficient as a Function of Reynolds Number.

In addition to the Reynolds number, there are four other important geometric tested parameters which affect the value of heat transfer coefficient of a single heated rectangular component, placed in an in-line array of rectangular components. The first parameter, H/t, indicates the fraction of the total flow that passes over the top of the components. Since most of the heat transfer takes place from the top of the component, the ratio H/t is an important parameter. The second parameter, S/L, expresses the flow disturbance due to interaction of outer flow and cavity flow formed by two neighboring components. The third parameter, t/L is associated with the total wetted surface area of the component exposed to the air flow. Furthermore, the location of a heated component (row number, r) in such an array can be considered as the fourth effective geometric parameter, since the upstream components have hydrodynamic effects on the heat dissipation rate of the downstream heated components. Effects of these geometric key parameters are discussed in detail in the following.

Effects of Heated Component Row Number (r)

Figure 5 shows variation of the heat transfer coefficient for different row numbers for the case of S/L = 0.43 and H/t = 1.25 with two different component heights and three different Reynold numbers. As seen from this figure, the heat transfer coefficients of the heated components along the first row are always the dominant values which are about 9 to 12% greater than those of the succeeding second rows. Figure 5 also conveys the fact that as the row number increases, the differences between h for the two neighboring rows become smaller. This difference reduces to the order of 1% for the fifth and sixth rows for the case of t/L = 1. Therefore, the heat transfer coefficient of the heated component for this case at the fifth and all subsequent rows is defined as the "periodically fully-developed" heat transfer coefficient.

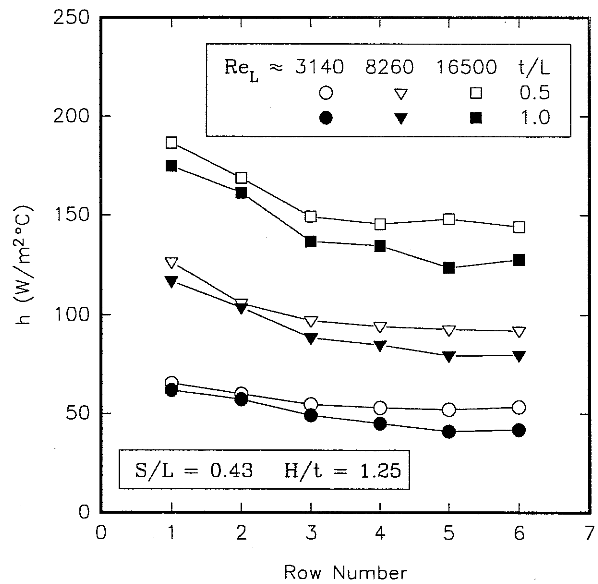

Figure 5. Heat Transfer Coefficient as a Function of Row Number.

However, Fig. 5 shows that the heated component of the case of t/L = 0.5 is a little less sensitive to the row number changes than the case of t/L = 1. Sparrow et al. (1982), found that it took five rows of components for the heat transfer to be truly "periodically fully-developed", while Wirtz and Dykshoorn (1984) reported three rows. However, they did not clearly mention what is the percentage difference between the heat transfer coefficient at the row they attained fully-developed, and its first downstream row. Heat transfer coefficient of the first row can be compared with fifth row (fully developed) in Fig. 4 for different channel heights for the case of t/L = 1 and S/L = 1 for different Reynolds numbers. It can be seen from this figure that the heat transfer for the first row is 9 to 12% greater than the corresponding fully-developed heat transfer coefficient. This finding is in good agreement with the results reported in the literature (Arvizu, 1981; Sparrow et al., 1982; Wirtz and Dykshoorn, 1984; etc) which indicates that the components along the first row of an in-line arrangement experience heat transfer which is 10 to 15% greater that those of succeeding rows.

Effects of Channel Height (H/t)

The dimensionless channel height, H/t, can be regarded either as a descriptor of the hardware or as a parameter in determining the air flow above the components. Figure 6 shows the behavior of the heat transfer coefficient changes owing to the changes of the channel heights (H/t) for different row numbers for two different S/L, fixed Reynolds number of 16700 and t/L = 1. As the channel height increases from 0.5 to 2.0, there is an average decrease of about 29% in the heat transfer coefficient for different row numbers and S/L. For a fixed approach velocity, as H/t increases, the portion of air flow over the component is decreased, thus reducing the heat transfer. This expression justifies the assumption of other investigators, that the heat transfer of a single component depends on the velocity around it. For example, see Moffat, et al. (1985).

Effects of t/L and S/L

Effects of these two important geometric parameters on the heat transfer coefficient of the heated component are clearly shown in Figs. 5 through 8. It can be seen from Figs. 5, 7, and 8 that as t/L is increased, the heat transfer coefficient decreases. This fact can be justified by defining a new-parameter called Surface Blocking Ratio (SBR). SBR is the ratio of the surface of the heated component blocked by its neighboring components in the direction of air flow, to the total surface of the heated component exposed to convection. This ratio is t/L/(1+4t/L) for the first and last rows, and 2t/L(1+4t/L) for the components in the other rows. Comparing this new defined parameter (SBR) for the two cases of t/L = 0.5 and 1.0, it reveals that for the case of t/L = 0.5, less fractional surface of the heated component is blocked by its neighboring components in the direction of the air flow (compare 1/3 to 2/5), which in turn causes more fractional surface exposed to convection, thus more convective heat transfer.

The results of this study showed that the more t/L effect becomes distinguished, the more array configuration becomes sparse, for example: with S/L = 1.0 and H/t = 1.25, as t/L is changed from 1.0 to 0.5, the average heat transfer coefficient through different Reynolds numbers increased by about 28%, from 70 to 90 W/m^2°C, while the increase was about 19% with S/L = 0.43, from 81 to 97 W/m^2°C (see Fig. 8). From these results, it can be concluded that the SBR effect is less pronounced with dense configurations because in this case the air flow paths between adjacent components become narrow. Thus, for the case with smaller S/L ratio the effect of SBR can be less pronounced.

The dimensionless inter-module space, S/L, equally deals both with streamwise and spanwise spacings. In viewing the streamwise direction, S/L expresses the flow disturbance due to the interaction of over-component flow and streamwise cavity flow formed by the two neighboring components according to Lehmann and Wirtz (1985) and Chang et al. (1987). These investigators found that the two neighboring components generate a streamwise cavity whose structure could have influence on the flow unsteadiness in the over-array layer. When the component spacing was large, strong interaction between the over-array layer flow and the cavity flow occurred and the level of unsteadiness was increased and the heat transfer from the component surface was enhanced with the same approach velocity. From the viewpoint of spanwise direction, S/L states inter-component paths of flow which traverse through the gaps between columns of components.

Figure 8 discloses the heat transfer coefficient variation through different Reynolds numbers with two different S/L. As S/L changed from 1.0 to 0.43, the heat transfer coefficient increased from 71 to 79 W/m^2°C with $Re_L \approx 8390$, H/t = 1.25, and t/L = 1.0 at row number 5. This increase in the heat transfer coefficient due to the changes of S/L was observed in most cases and can be explained as follows: as the sparse array is changed to the dense array, the spanwise flow path area is reduced and in turn the array velocity becomes faster. This increased array velocity contributes to more convective heat transfer to the array component surfaces, while the reduced streamwise and spanwise inter-component paths present less convective heat transfer.

Correlation

The systematic experimental data obtained in this study were used for development of a general empirical heat transfer correlation. It is well known that the heat transfer data for internal flows and a specific fluid can be correlated by the equation of the form h or Nu = c_g (Re)^n, where the geometric coefficient c_g and the exponent n are empirically determined constants depending on the flow and geometric conditions. The heat transfer results reported in the open literature indicate the bounds on the exponent, n, to be 0.5 to 0.8. It appears from the reported correlations that one may find an exponent for the Reynolds number depending on the test array density and, furthermore, the exponent can be a function of array inter-component spacing. The upper limit on the exponent, i.e., n = 0.8 is the value reported for turbulent flow through smooth parallel planes (Kays and Crawford, 1980). As the regular in-line array of rectangular components become very dense, i.e., S/L << 1, the exponent n increases and approaches the behavior of smooth planes.

Effects of Reynolds number and four geometric key parameters on the heat transfer coefficient were correlated by:

$$h = 0.208\,(Re_L)^a\,(R_L)^b\,(\Delta)^{-0.841}\,(t/L)^{-0.141} \qquad (4)$$

where

$$a = 0.44 + (S/L)\exp[-1.639(S/L)],$$
$$b = -0.052(S/L)^{-0.835}$$

$$2765 \le Re_L \le 17230,\ 0.5 \le H/t \le 2.0,$$
$$0.5 \le t/L \le 1.0,\ 0.43 \le S/L \le 1.0$$

$$R_L = (r - 1)(1 + S/L) + 1/2$$

and

$$\Delta = \frac{(H+t)(S+L) - tL}{(H+t)(S+L)}$$

where r is the heated component row number. Note that the exponent on Δ for special case of H/t = 2.0, t/L = 0.5, and S/L = 0.43 should be changed from –0.841 to –0.256. The R_L is the ratio of the distance between the leading edge of the first row component and the center of the heated

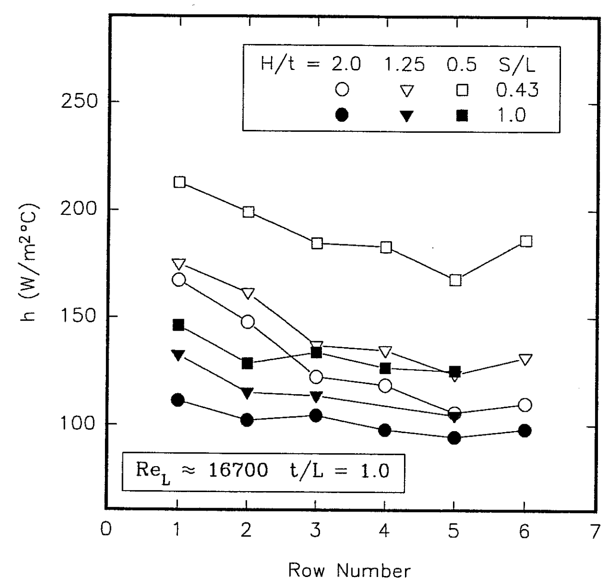

Figure 6. Heat Transfer Coefficient as a Function of Row Number.

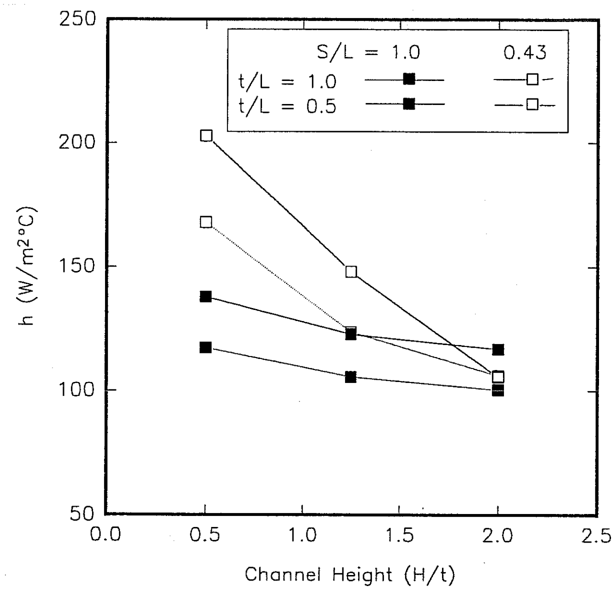

Figure 7. Heat Transfer Coefficient at Row 5 as a Function of Channel Height for the Case of $Re_L \approx 16400$.

component, to the characteristic component length, L. The Δ is defined as the ratio of the frontal streamwise flow path area to the total frontal surface area. Since the use of half components at the side columns is to simulate an infinitely wide array model more closely (Sparrow et al., 1982), non-duplicating frontal area is used to calculate the Δ value.

Equation (4) has the array inter-component spacing (S/L) relations with the exponent on the Reynolds number and the exponent on the row number variable (R_L). As discussed earlier, the decrease in the heat transfer

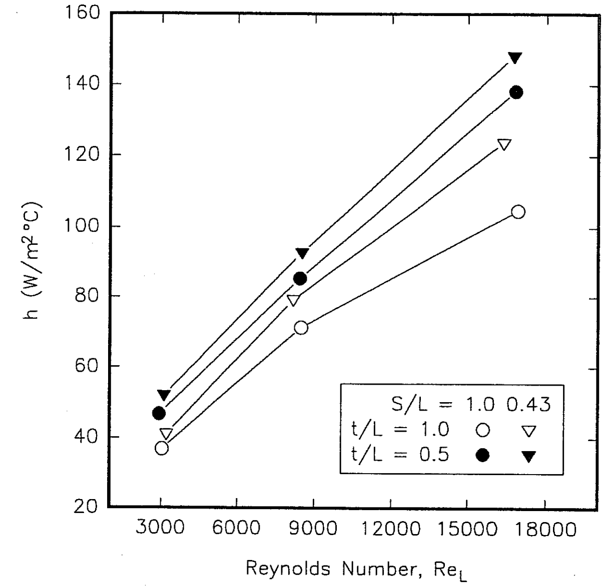

Figure 8. Heat Transfer Coefficient at Row 5 as a Function of Reynolds Number for the Case of H/t = 1.25.

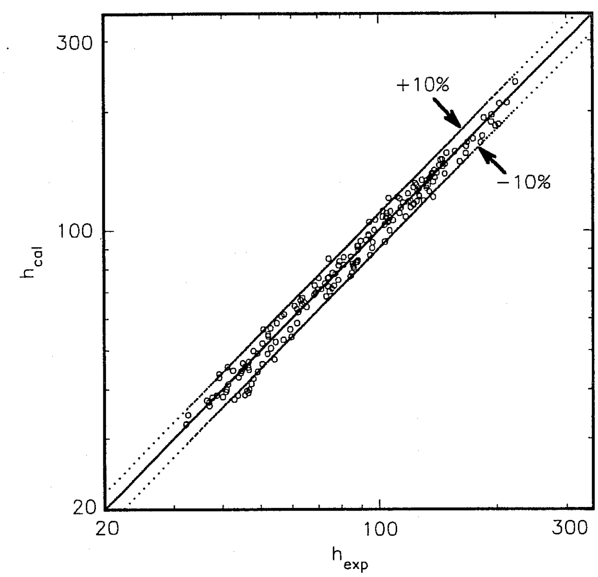

Figure 9. Comparison of the Predicted Heat Transfer Results with the Experimental Data.

coefficient with row numbers are different at different inter-component spacings, S/L. The dense array layout (S/L = 0.43) has around two-fold more heat transfer coefficient variation due to the row number changes than the sparse array layout (S/L = 1.0).

Equation (4) is applicable to any single heated component in an in-line array of similar rectangular components having different geometries and approach velocities within the above reported ranges. Figure 9 reveals the deviation between the experimental heat transfer coefficient data and the results predicted by Eq. (4).

Table 2. Comparison of Equation (4) with Other Works

Study	Ranges of Comparable Tested Experimental Parameters					%Deviation
	Re_L	Row No.	H/t	t/L	S/L	
Sparrow et al., 1982 (L = 26.7 mm)	2000 to 7000	5	1.67	0.37	0.25	Mean 33 % -41 to -22%
Wirtz and Dykshoorn, 1984 (L = 25.4mm)	1300 to 13000	3	1.25	0.25	1.0	Mean 11% +14 to +8%
Copeland, 1992 (L = 37 mm)	5000 to 13000	5	2.0	0.16	0.37	Mean 37% +38 to +33%
					1.05	Mean 34% +35 to +32%

The correlation gives a representation of the experimental data to within +15.2% and –14.6%. In the development of the correlation, a total of 179 experimental data points were used. The absolute average deviation between the results predicted by Eq. (4) and the experimental data is 4.7%. About 60% of the data (106 data points) were predicted with less than ±5% deviation and only 8% of the data (15 data points) were predicted more than ±10% deviation.

In order to verify the accuracy of Eq. (4), a set of separate experiments were performed with different geometric parameters of t/L = 0.75, S/L = 0.67, and H/t = 1.25 and with the same range of the Reynolds number. These channel height, component height, and inter-component spacing are the middle values within the ranges reported for Eq. (4). These new experimental data obtained for the intermediate geometric parameters were compared with Eq. (4). The experimental data were predicted with less than ±10% deviation which is well within the correlation maximum percent deviation. This good agreement between the predicted results and the experimental data show the general applicability of the proposed correlation within its own range of geometric parameters (H/t, t/L, and S/L).

Comparisons

Equation (4) was compared with the correlations of other investigators which have different geometries (Table 2). Reynolds numbers of their studies were based on the characteristic component length, L, and Eq. (4) was simplified for each case according to their "Ranges of Comparable Tested Experimental Parameters" as shown in Table 2.

A good agreement between the results predicted by Eq. (4) and the correlation of Wirtz and Dykshoorn (1984)was obtained. The highest percent difference is within Eq. (4) error band. The proposed correlation predicts a little higher heat transfer coefficients and slightly stronger dependence on the Reynolds number. From this comparison it can be concluded that the proposed correlation can be used beyond its recommend range of t/L, since the only difference between the recommended parameter ranges for both correlations is the t/L range. They used "flatpacks" (t/L = 0.25), while the t/L range for the correlation of this study was 0.5 ≤ t/L ≤1.0.

The proposed heat transfer correlation was used to predict the experimental results of Sparrow, et al. (1982) for their reported ranges of tested experimental parameters. The correlation predicted lower heat transfer coefficients and a weaker dependence on the flow rate. The reason for the differences between the correlation's prediction and Sparrow, et al's could be that their array is more densely packed (S/L = 0.25) and the exponent on the Reynolds number is 0.72, whereas this study's packing density range is from 0.43 to 1.0 and the exponent of the Reynolds number in Eq. (4) is 0.61 for S/L = 0.25, implying that the correlation of this study does not provide the proper prediction exponent number of the Reynolds number beyond its recommended packing density range. Furthermore, they used smaller component height (t/L = 0.37). From this comparison, it can be concluded that the proposed correlation should not be used beyond its recommended range of S/L.

Copeland (1992) used very "flatpack" arrays (t/L = 0.15) but nearly the same packing densities as this study. This study's correlation predicted higher heat transfer coefficients and a little stronger dependence on the Reynolds number, implying that the experimental heat transfer coefficient changes due to the element height can not be the monotone variation assumed by the correlation. It can be concluded from this comparison that the proposed correlation should not be used for the very flatpack arrays like t/L = 0.16.

CONCLUSIONS

The results of this study appears to be the first systematic experimental investigation of the effects of all relevant geometric parameters and air flow rate on the heat transfer coefficient of a single heated rectangular component placed in an in-line array of unheated similar rectangular components. The experimental data collected in this study were used to develop a general empirical heat transfer correlation. This correlation is of significance in electronic cooling and has sufficient generality in order to be "transportable" to any in-line array of rectangular components having arbitrary geometries i.e., different H/t, t/L, S/L, and r within the reported ranges. It can be used by industrial designer as a tool in order to predict the self-temperature rise of any heated rectangular component (due to its internal heat) placed in such array. Results of this study revealed important specific conclusions:

1. In general for a fixed Reynolds number, the increase in any of the three geometric parameters (H/t, t/L, and row number) caused decrease in the heat transfer coefficient of the heated component. The results of this study showed that this fact is true for the fourth parameter; i.e, S/L, but as S/L increases the heat transfer coefficient becomes less sensitive to it. It is believed that if more experiments were performed for a range of S/L, some critical value for S/L can be found that for S/L greater than this critical value the trend will reverse. It was found that due to the negative exponent for t/L and H/t, the heat transfer coefficient is more sensitive to the values of t/L < 1 and H/t < 1. Thus, if

more experiments for ranges of t/L > 1 and H/t > 1 were performed, some critical values for these two parameters can be found that beyond these critical values the variation of heat transfer coefficient will be negligible. It was also found that the effects of geometric parameters and Reynolds number on the heat transfer coefficient of the heated component were not independent of each other, but they were inter-related and have combined effects. For example, exponents of Reynolds number and row number in Eq. (4) were functions of S/L, while exponent of t/L and H/t were related to each other.

2. Equation (4) is a complement to the superposition correlation found by other investigators (see Arvizu, 1981; Arvizu and Moffat, 1982; Anderson and Moffat, 1990; etc.) for thermal wake of the component due to its upstream heated component(s). With the help of Eq. (4) and any superposition correlation within the range of Eq. (4), it is possible to predict the operating temperature of any component in an in-line array of similar rectangular component.

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