Forced Convection Heat Transfer from a Flat Plate at Different Locations between Plates Array.

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FORCED CONVECTION HEAT TRANSFER FROM A FLAT PLATE
AT DIFFERENT LOCATIONS BETWEEN PLATES ARRAY

انتقال الحرارة بالحمل الجبري من لوح مستوي عند مواقع مختلفة بين صفيف الواح

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خلاصة :

هذا العمل جزء من برنامج مستمر خاص بفهم أفضل حدود استخدام اضواء كمرود للأجهزة الالكترونية . وكذلك تطوير الطرق التي تمكن من نوسيع مجال استخدام كمرود هذه الأجهزة . بتضمن هذا البحث دراسة عملية لانتقال الحرارة بالحمل القسري من لوح مستطيل سنوي ساخن مصنوع من الألومنيوم المصقول (١٥٥ x ١١٥مم) الى تيار هواء . اللوح الساخن موضوع بين صفيف من الاتواح الغير مسخنة حيث يأخذ أوضاع مختلفة . بين صفيف الالواح ، في اتجاه انسياب تيار الهواء . ويتعلق هذا البحث أساسا بدراسة تأثير موقع اللوح المسخن بين صفيف الالواح على معامل انتقال الحرارة المتوسط . ولقد تم تصميم وانشاء جهاز تجريبي لغرض القيام بهذه الدراسة . ولقد غمت التجارب مواقع اللوح المسخن والتي تبعد عن الجدار الأيمن الرأسي لقناة النفق الهوائي المستخدم (١٥٥ x ١٢٥مم),بمسافات ١٢,٥ ، ٢٢,٥ ، ٣٢,٥ ، ٤٢,٥ و ٦٢,٥ مم . وأثناء الدراسة تم تغيير رقم ريبولدز من ٦,٦٨٠ الى ١٠٧,١٧٠ . وتوضع الدراسة مدى تأثر رقم نوسلت على موقع اللوح المسخن بين صفيف الالواح الغير مسخنة . ولقد أجريت التجارب على اللوح المسخن في حالة استخدام فيض حراري منتضم . وتم صياغة علاقة بين رقم نوسلت وكل من رقم رينولدز والموقع اللابعدي للوح الساخن . ولقد عملت دراسة مقارنة بين نتائج هذا البحث والأبحاث المتاحة في اديبات موضوع البحث.

ABSTRACT

A forced convection heat transfer experiment was conducted to study the heat transfer from hot rectangular flat plate (155 x 115 mm) to an air stream. The hot plate was situated between unheated plates array at various locations in the flow direction. The study was mainly concerned with the effect of the plate location on the average film coefficient of heat transfer . An experimental test rig was designed and constructed for this investigation. The experiments covered the plate locations of x = 12.5, 22.5, 32.5, 42.5 and 62.5 mm, and extended over a range of Reynolds number from about 6,680 to 107,170. A correlation between the average Nusselt number, the Reynolds number and the hot plate dimensionless location (X) was deduced. The experiments were performed for the case of uniform heat flux. A comparison with the available work in the literature review was also made.

INTRODUCTION

Our previous study [1] on the heat transfer from a flat plate that situated at different locations in the wind tunnel channel presented a formula for the average Nusselt number. The flat plate was located vertically in the direction of air flow stream. The obtained formula does not indicate that the plate location has a significant effect on the average heat transfer coefficient. The obtained correlation is in the form:

$$Nu = 1.97 \ Pr^{0.33} \ Re^{0.44} \qquad (1)$$

where $\qquad 4,300 \le Re \le 67,000$

This paper presents the results of a subsequent investigation conducted to establish a formula for the average Nusselt number of the hot plate situated, in the wind tunnel duct, at different locations between number of unheated plates array.

Recent advances in semiconductor technology have led to increasing miniaturization in circuit design. As electronic components are made smaller, the amount of heat that must be dissipated per unit volume of a device increases dramatically. For example, the peak heat fluxes in a recent computer circuit are 10 times greater than those in an older computer [2]. Forced convection cooling of the circuitry is commonly used to maintain the desired operating temperature in such a device. The present work is part of a continuing program to understand better the limits of air cooling, and to develop methods for extending its usable range. However, in the design of finned heat exchanger surfaces and electronics packages is how to choose the best location of heat generating plates. Several investigators have attempted to increase the rate of heat transfer in a circuit board[3-9]. Shalaby, et. al,[10-12] studied forced convection heat transfer at an inclined and yawed rectangular flat plate. They studied the tripping wires effect on heat transfer during air flow over the test plate.

The recent reviews of the fundamental heat transfer literature on electronic equipment cooling [13,14] show that the best location and the optimal spacing have not been determined for packages that are cooled by forced convection. The present study concerned with this problem, and develops concrete means for finding the best location of the heated flat plate situated in a wind tunnel channel between unheated plates array, in the case of laminar flow. The problem considered is that of air flow through array of unheated flat plates located in the wind tunnel duct. The heated flat plate, 115 mm x 155 mm, is situated between the flat plates array. During the course of the experimental work the heated plate location changes to have lateral distance (x = 12.5, 22.5, 32.5, 42.5 and 62.5 mm) between its center and right inner vertical-wall-face of the duct. The wind tunnel duct has a squared cross-section of 125 mm x 125 mm.

EXPERIMENTAL APPARATUS AND PROCEDURE

An experimental test rig is designed and constructed for the planned experiments as shown in Fig.(1). It consisted of a low-turbulence open circuit wind tunnel of a 125 mm square cross section, in which air from the laboratory room is drawn through the system by a downstream blower (13) after passing through a 150-cm long Plexiglas entrance section. The flow rate is controlled by a throttle valve (15). A flexible textile plastic connection (12) is fitted between the blower and the duct to prevent any vibration transmission to the honey comb section (11), the main duct (6) and the test section (8). The velocity of the air stream drawn through the main duct is measured with the help of Pitot tube (18) and an inclined alcohol manometer (19) at the centers of nine imaginary equal areas into which its plane was divided. The Pitot tube is situated 50 cm upstream. The air velocity is, also, measured by the hot-wire probe(3) located 75 cm upstream. The difference in velocity values measured by the two methods is about $\pm$ 1.0%. A general layout of the experimental apparatus is illustrated in Fig.(1-a) with the associated air supply system and the heating plate.
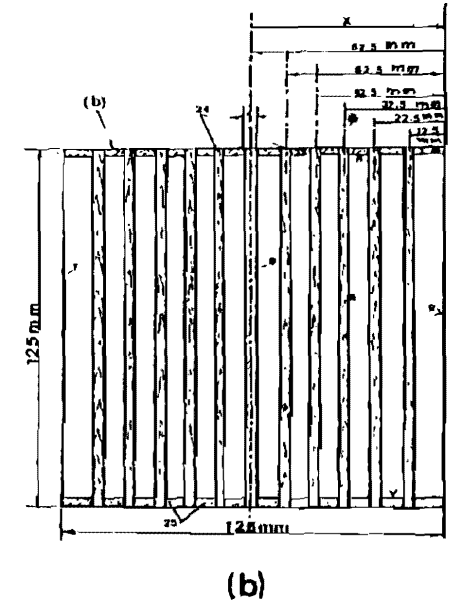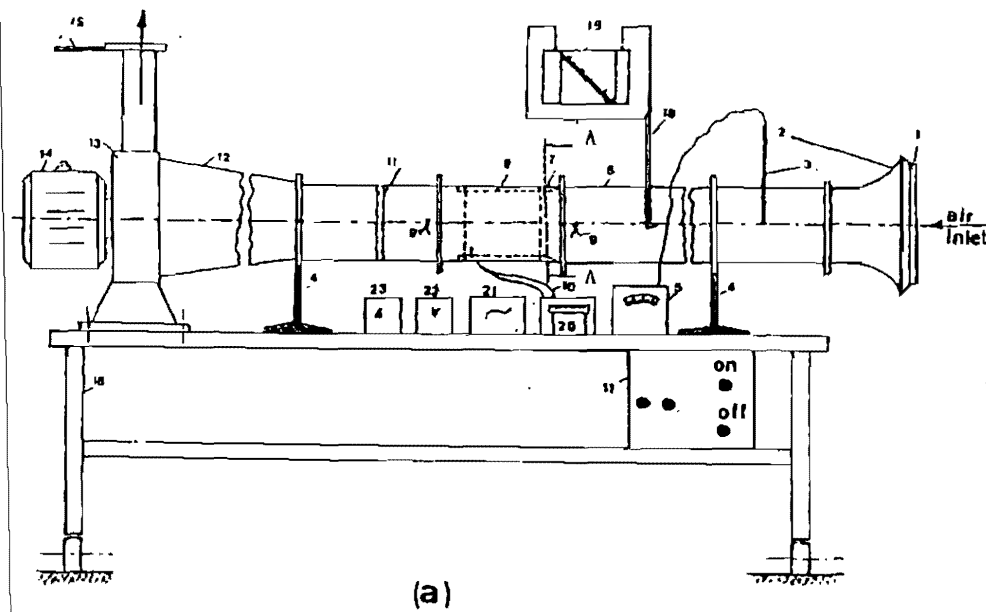
(a)

(b)

Section at A-A.

Fig.(1) Experimental test rig.

1-stabilizing plate, 2-collector, 3-hot wire anemometer probe, 4-carriers, 5-hot wire anemometer, 6- straight channel, 7-test section channel, 8-tested plate, 9-copper-constantan thermocouple, 10-thermocouples connection, 11-honeycomb, 12- connection part, 13- down stream blower, 14-electric motor, 15-control discharge gate, 16-movable table, 17-electric panel, 18-Pitot tube, 19-inclined manometer, 20-temperature recorder, 21-autotransfomer of the test plate heater, 22-voltammeter, 23-ammeter, 24-wooden plates, 25-separation wooden slapes.

Figure (1-b) shows that the heated test plate (8) is located vertically, between an array of unheated wooden plates (24), in the direction of the main flow stream. The array plates are fixed in their locations by the help of wooden slots (25). These slots are fixed on the top and bottom of the wind tunnel channel. The front of the wooden slots has a triangular shape with 15 deg, on the horizontal surfaces of the channel, to avoid the air disturbance motion near the test plate. Each wooden plate of the array wooden plates has the same outside dimensions of the hot flat plate. The test plate is located far from the right vertical wall of the wind tunnel channel by five different distances x = 62.5, 42.5, 32.5, 22.5, and 12.5 mm. The distance between the array plates' center equals 10 mm.

Fig.(2) shows a schematic diagram of the test plate. It has 155 mm length 115 mm width and 5 mm thickness. The faces of the test plate are made from two polished aluminum sheets (1), each of them has 1.0 mm thickness. The back surface area of each face is divided to nine imaginary equal rectangular areas, each of 52 mm length and 38 mm width approximately. In the center of each area a copper-constantan thermocouple (5), made from 30 gauge wires, is fixed in thin slots (0.5 mm deep) cut on the under side of the plate. The calibration of thermocouples was performed and the calibration curve was drawn from which, the average of the nine local temperatures is obtained. The temperatures are sensed at a depth of about 0.5 mm from the top polished surface. As such, these values can be taken to represent the outer surface temperatures because of the use of the aluminium plate in which the difference between the estimated outer surface temperatures and the measured values are found to be, in general, less than 0.03 C. The plate is heated electrically by means of the heater(3), which consisted of nickel-chromium heating wire wound around a threaded sheet of mica (4) and sandwiched ,also, between two sheets of mica (2). Each mica sheet has 153 mm length, 113 mm width and 0.5 mm thickness. The leading edge of the plate is rounded in the direction of flow to avoid disturbance in the trailing edge. The average of the nine local temperatures is obtained for each face of the test section plate as follows:

$$T_w = \sum_{n=1}^{9} T_n/9 \qquad\qquad (2)$$

where n is the location number of the measured point.

The test plate sides are insulated by three layers of glass wool tape. On the outer surface of the tape, eight thermocouples are placed and embedded at the mid-height of the surrounding tape (6). The purpose of these thermocouples is to determine the heat loss by conduction (Q_{cond}) from the test plate sides. The mean bulk air temperatures far from the two sides of the plate are measured using two movable thermocouples facing each side. All the thermocouples are connected to a six-point-temperature recorder ((20)-Fig.(1)). The temperature of the duct walls is measured by three thermocouples fixed on the right, left and upper walls of the wind tunnel channel, at the working section. The dry bulb temperatures at the intake of the air blower and at the outlet of the working section are measured.

The heat input to the test plate is controlled by using an auto-transformer (21) as well as one voltammeter (22) and ammeter (23),as
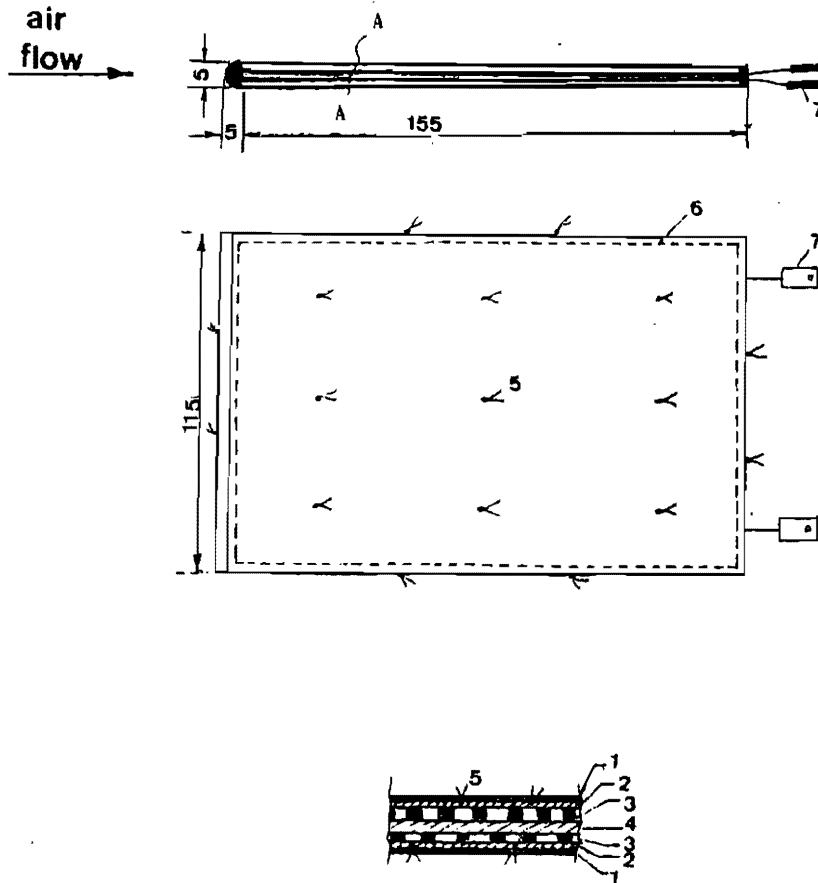
Section at A-A

Fig.(2) Construction of the hot plate.

1-two aluminum plates of 1.0 mm thickness, 2-two plates of mica with thickness of 0.5 mm, 3-rectangular cross section electric heater, 4-mica plate of 0.5 mm thickness, 5-copper-constantan thermocouples, 6-glass wool insulation, 7-electric heater connection.

seen in Fig.(1). Once the switch of the power source closes, the heating system starts. During the course of this experimental work nearly 2.5 hours are needed to reach the steady state condition. This condition is satisfied when the temperature reading does not record any change within a time period of about 15 minutes. Applying the principle of conservation of energy gives:

$$Q_{in} = Q_{cond} + Q_{conv} + Q_{rad} \tag{3}$$

As the power leads, and the thermocouples wires have very small cross sections and the various connections in the apparatus are made of insulated materials, then, the amount of heat conducted away through these members is very small and, therefore, can be neglected.

In order to find the heat lost by radiation (Q_{rad}) the average value of the emissivity of 0.24 for polished aluminium plate was taken from [15,16], in which they reported that no significant dependence of emissivity on temperature was observed. The estimated values of Q_{cond} and those of Q_{rad} were of order 15 to 35% of the input power to the hot plate heater. The average convective heat transfer coefficient is determined from the following expression;

$$h = (Q_{in} - Q_{cond} - Q_{rad})/\{ A(T_w - T_a)\} \tag{4}$$

The probable error in finding the average heat transfer coefficient was estimated to be about $\pm$ 7%. The projected area of the test plate on a vertical plane perpendicular to the tunnel axis equals the cross-sectional area of the test plate (the plate width x the plate thickness). It is found that if the blockage of the wind tunnel free stream cross-section area is about 10%, Test and Lessmann[15] have reported that their heat transfer results with and without blockage differ by a maximum of 7%. In the present work the blockage of the wind tunnel free stream cross-section area is about 44%. Therefore, the blockage may affect the heat transfer results to some extent.

RESULTS AND DISCUSSION

In order to find the average heat transfer coefficients and their correlation with air flow, some quantities are measured for each data run. The power input to the test plate heater, the rate of heat lost by radiation from the plate surfaces to the wind tunnel channel walls, the heat lost by conduction from the test plate sides, the average flat plate surface temperature, the air flow stream velocity and the free stream temperature are recorded. The net rate of heat transfer by convection is used to calculate the average heat transfer coefficient from equation (4). The characteristic length used in both Nusselt number and Reynolds number is the test plate length (L). The air density is taken at the bulk temperature, while the other properties are taken at the mean film temperature.

The parameters that are varied independently during the course of experiments included the main air stream velocity (or Re) and the test plate location (x). The Reynolds number spanned the range between 6,680 and 107,170 and the values of x chosen are 12.5, 22.5, 32.5, 12.5 and 62.5 mm. The results for different locations could be represented by equation (5) and Table (1) as follows:

$$Nu = c\ Re^m \tag{5}$$

where c and m are constants shown in Table (1).

Table (1): Values of the constants of heat transfer equation (5)

Plate location (x) in mm	Values of the constants	
	c	m
12.5	0.372	0.58
22.5	0.428	0.56
32.5	0.469	0.55
42.5	0.501	0.54
62.5	0.573	0.53

The results of different plate locations are plotted in the form of Nusselt number versus Reynolds number, as shown in Fig.(3-7). Each figure is drawn at a certain value of x. The figures, also, have the dashed line obtained from correlation (1) as well as the solid line obtained from the following correlation[17];

$$Nu = 0.6795 \ Pr^{0.33} \ Re^{0.5} \tag{6}$$

where $Re \leq 3 \times 10^5$

This correlation has been obtained by using integral method in the case of laminar heat transfer and with a constant-heat flux from the test plate surface .

Results of Nu values versus Re, in case of x = 12.5 mm, are shown in Fig.(3). Generally, one can see that Nu value increases with Re. Also, it can be observed that, the Nu values obtained from the present experimental work are, generally, higher than the solid line of equation (6) and less than the dashed line of equation (1). This may be because of the blockage effect. It is also seen that the slop of the present results line is higher than the slops of the two lines obtained by equations (1) and (6). The present data show some approach with the same obtained by equation (6) at law Re values and show a quite good approach with the data obtained by equation (1) at high Re values . This may be because in the case of air flow through a rectangular duct (like the air flow through the plates array) the coefficient of friction decreases with the increase of Reynolds number [18].

Results of the Nu values versus Re values in the case of x equals 22.5, 32.5, 42.5 and 62.5 mm, are illustrated in Figs. (4, 5, 6 and 7) respectively. The main observation, comes out of these four figures, is the slop of the Nu values versus Re values decrease with the increase of x, i.e. as the test plate location moves far from vertical channel wall. This may be because of the shape of the air flow velocity profile at the plates array entrance. In the case of fully developed laminar flow, the air velocity value increases with x to have the maximum value at the duct center.

In all, 132 data points obtained are represented in Fig. (8). One may clearly observe that, the Nu values, generally, increases with Re. The data shows a dependence on the test plate location (x). Along
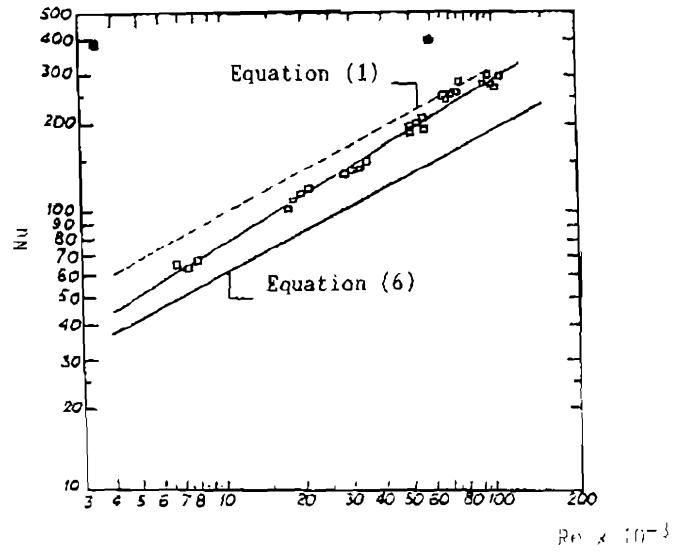
Fig.(3) Average Nusselt number versus Reynolds number ,
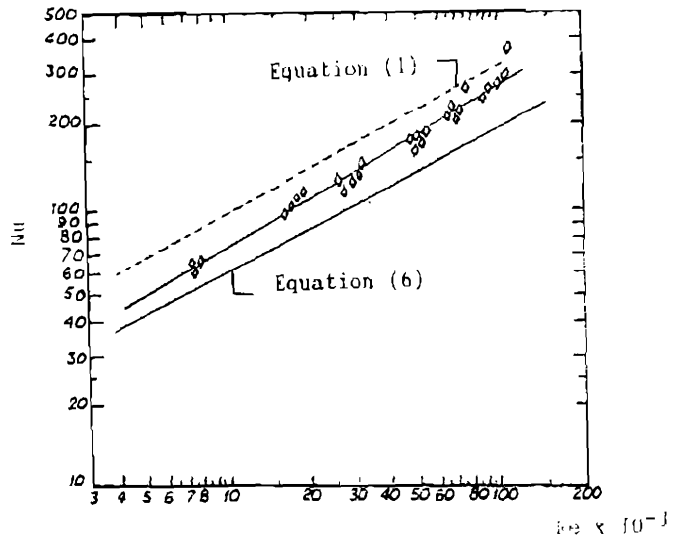at x= 12.5 mm.

Fig.(4) Average Nusselt number versus Reynolds number ,
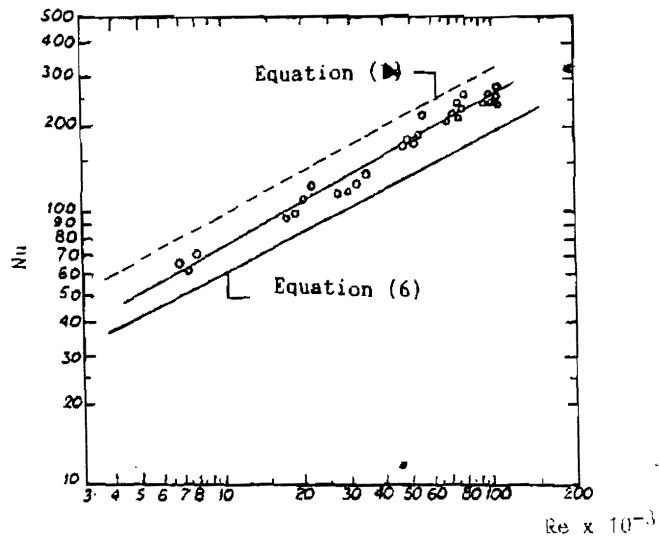at x= 22.5 mm.

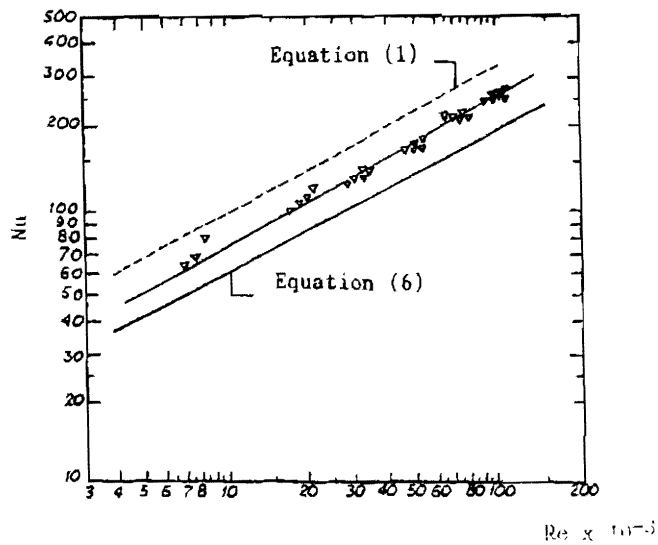Fig.(5) Average Nusselt number versus Reynolds number ,
at x= 32.5 mm.

Fig.(6) Average Nusselt number versus Reynolds number ,
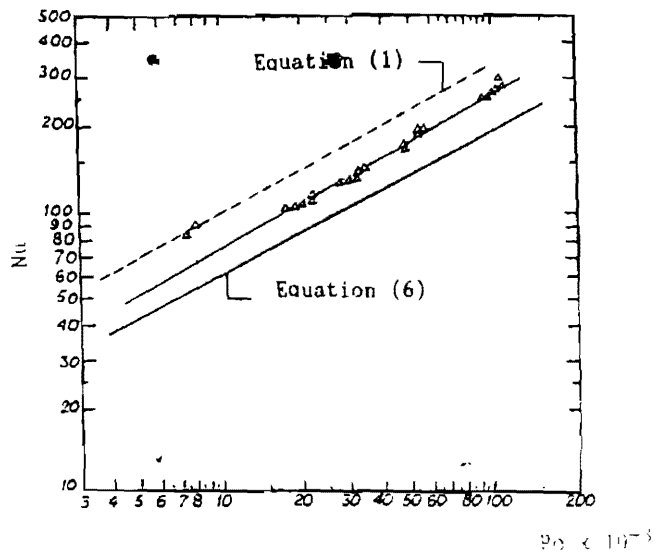at x= 42.5 mm.

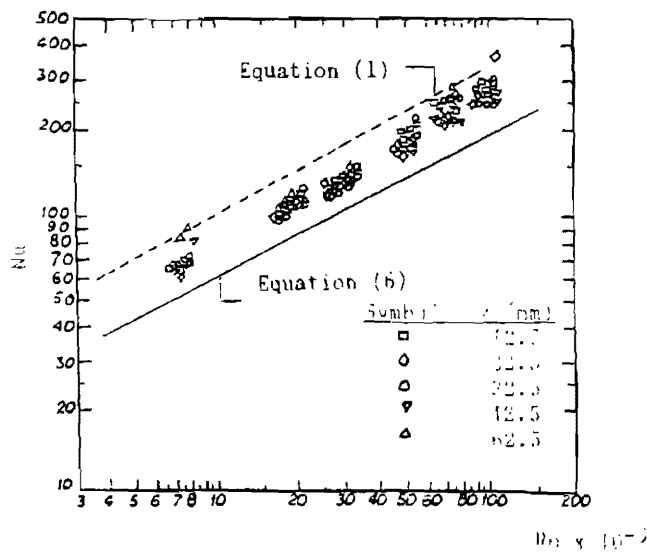Fig.(7) Average Nusselt number versus Reynolds number ,
at x= 62.5 mm.

Fig.(8) Average Nusselt number versus Reynolds number ,
at different locations.

the course of this work the heat input to the test plate heater
varies between 3.70 W and 185.5 W.

Figure (9) shows Nu values versus x at Re equals 100,000, 50,000,
20,000 and 7,000. One may observed that the Nu values decrease with
the increase of x, i.e. as the test plate moves toward the channel
center. At high Reynolds number the test plate location (x) affected
more the Nusselt number values than in the case of lower Re values,
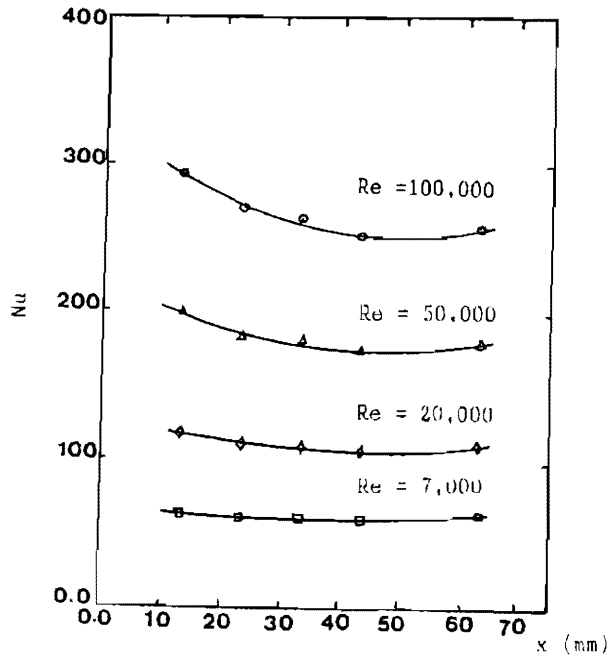this may be because of the above notation comes from [18].

Fig.(9) Average Nusselt number versus not plate location.

General Heat Transfer Correlation:

As shown in Fig.(8), the results of the plate locations 12.5 mm to
62.5 mm may be taken as one group. A trial was made to find out a
correlation between Nu and Re and the dimensionless distance far from
the right wall of the wind tunnel channel(X = 2x/b). The obtained
correlation was in the form:

$$Nu = 0.573 \ X^{0.29} \ Re^{0.53} X^{-0.055} \qquad (7)$$

where $0.2 \leq X \leq 1.0$ and $6,680 \leq Re \leq 107,170$

One may observe that the dimensionless distance X appears in the
obtained correlation, i.e. the Nu values depend on the test plate
location (X) between the plates array, as previously observed. The
experimental data, shown in Fig.(8), agree with the values obtained
from correlation (7) by ± 10%.

CONCLUSIONS

As a result of the present investigation the following conclusions are derived:

1- As the Reynolds number increases the average Nusselt number value increases, i.e., the average heat transfer coefficient increases with Reynolds number.

2- The obtained correlation indicates that the test plate location between the plates array has a significant effect on the average heat transfer coefficient, specially at high Reynolds number values.

3- The test plate location does not affect much the average heat transfer coefficient, at low Reynolds number values.

NOMENCLATURE

The symbols used in the paper have the following meanings:

A = Plate surface area, [m^2],
b = Wind tunnel channel width, [m],
Cp = Specific heat of the fluid, [kJ/kg.K]
h = Average heat transfer coefficient, [$W/m^2 \cdot K$],
k = Thermal conductivity of the fluid, [W/m.K],
L = Test plate length, [m],
Nu = Average Nusselt number, [h.L/k],
Pr = Prandtl number, [μ Cp/k],
Q_{cond} = Rate of heat lost by conduction, [W],
Q_{conv} = Rate of heat lost by convection, [W],
Q_{in} = Power input to the test plate heater, [W],
Q_{rad} = Rate of heat lost by radiation, [W],
Re = Reynolds number, [$u_{av}.L/\nu$],
T_n = Test plate surface local temperature, [C],
T_w = Test plate surface average temperature, [C],
T_{α} = Ambient temperature, [C],
u_{av} = Axial average velocity component, [m/s],
X = Dimensionless distance from the right hand side wall of the wind tunnel channel, [2x/b],
x = Distance from the right hand side wall of the wind tunnel channel, [m],
μ = Dynamic viscosity of the fluid, [$N.s/m^2$],
ν = Kinematic viscosity of the fluid, [m^2/s],

REFERENCES

1-Shalaby, M. A. " Forced convection heat transfer from a flat plate at different locations", Accepted to be presented in First Engineering Conference- Mansoura, 28-30 March (1995).

2-Hwang, U. P. and K. P. Moran, " Boiling heat transfer of silicon integrated circuits chip mounted on a substrate", Heat Transfer in Electronic Equipment (edited by M. D. Kelleher and M. M. Yovanovich). American Society of Mechanical Engineers, New York (1984).

3-Bar-Cohen, A., and W.M. Rohsenow, " Thermally optimum spacing of vertical, natural convection cooled, parallel plates", J. Heat Transfer Vol. 106, pp.116-123,(1984).

4-Bejan, A. " Convection heat transfer", p.157, Problem 11. Wiley, New York (1984); Solution manual for convection heat transfer, pp.93-95, Wiley, New York (1984).

5-Sugawara, S., T. Sato, H. Komatsu and H. Osaka, "Effect of free stream turbulence on flat plate heat transfer", Int. J. Heat Mass Transfer. Vol. 31, No. 1,pp.5-12,(1988).

6-Sparrow. E. M., J. E. Niethammer and A. Chaboki, "Heat transfer and pressure drop characteristics of arrays of rectangular modules encountered in electronic equipment", Int. J. Heat Mass Transfer, Vol. 25, No.7, pp.961-973,(1982).

7-Sparrow, E. M., S. B. Vemur and D. S. Kadle, "Enhanced and local heat transfer, pressure drop, and flow visualization for arrays of block-like electronic components", Int. J. Heat Mass Transfer, Vol. 26, No. 5. pp. 689-699, (1983).

8-Moffar, R. I., and A. M. Anderson, "Applying heat transfer coefficient data to electronics cooling", ASME Jr. Heat Transfer, Vol.112, pp.882-890,(1990).

9-Anderson, A. M., and R. I. Moffar, "Direct air cooling of electronic components: Reducing component temperatures by controlled thermal mixing", Trans. ASME, Vol. 113, Feb.(1991).

10-Shalaby, M. A., F. F. Araid and A. A. Desoky, "Forced convection heat transfer at an inclined and yawed rectangular plate", Mansoura Bulletin Vol. 11, No. 1, June (1986).

11-Shalaby, M.A., and F. F. Araid, "Tripping wires effect on heat transfer during wind flow over rectangular inclined and yawed flat plate", M. B., Vol.11, No.2. December(1986).

12-Shalaby, M.A., and F. F. Araid, "Heat transfer and flow visualization of separated reattached air flow over reversed rectangular flat plate", MEJ Vol.12, No.1, June (1987).

13-Incropera, F. P. "Convection heat transfer in electronic equipment cooling", J. Heat Transfer Vol.110, pp.1097-1111,(1988).

14-Peterson, G. P., and A. Ortega, "Thermal control of electronic equipment and devices", Adv. Heat Transfer, Vol. 20, pp.181-314,(1990).

15-Test, F.L., and R. C. Lessmann, "An experimental study of heat transfer during forced convection over a rectangular body", ASME Jr. of Heat Transfer, Vol. 102, pp. 146-151,(1980).

16-Motwani, D. G., U.N. Gaitonde and S.P. Sukhatme," Heat transfer from rectangular plates inclined at different angles of attack and yaw to an air stream", Jr. of Heat Transfer, Trans. ASME, Vol. 107,p.307 ,(1985).

17-Holman, J.P., "Heat transfer", McGraw-Hill Book Co., p.232,(1989).

18-Kays, W. M., "Convective heat and mass transfer", TATA McGraw-Hill Publishing Company LTD., New Delh, p.58, (1976).