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FORCED CONVECTION HEAT TRANSFER FROM A FLAT PLATE
BETWEEN PLATES ARRAY WITH LATERAL AIR FLOW

انتقال الحرارة بالحمل الجبري من لوح مستوي بين صفيح اللوح

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خلاصة

يقدم البحث دراسة معملياً لانتقال الحرارة بالحمل القسري من لوح مستطيل مسخن الى تيار هوائي. والبحث كجزء من برنامج مستمر خاص بدراسة انساب حدود لاستخدام الهواء كمبرد في بعض الأجهزة الإلكترونية. واللوح مصنوع من الألمنيوم المصقول (155 X 115 مم) وموضوع بين صفيح من اللوح مشابهة غير مسخنة. يدخل الهواء الى النفق الهوائي المثبت به صفيح الألواح عمودياً من فتحة جانبية مستطيلة أو من فتحتين مع تغير وضع اللوح بمسافات 12,5, 22,5, 32,5, 42,5, 62,5 مم من حدار انجرى الهوائي. النفق الهوائي مقطعه مربع الشكل وأبعاده الداخلة 125 x 125 مم. وانشاء التجارب تم تغير سرعة الهواء حيث ترواحت قيمة رقم رينولدز بين 6800 و 109000 تقريباً. وقد تم أستنتاج بعض العلاقات اللابعدية لحساب معامل انتقال الحرارة بالحمل و تم مقارنة نتائج البحث مع بعض النتائج المتاحة في هذا الإتجاه.

ABSTRACT

An experimental work on heat transfer by forced convection was conducted in order 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 air flow stream enters the tunnel duct laterally from one side and from two sides. The study was mainly concerned with the effect of the lateral air flow 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 6,800 to 109,000. Two correlations between the average Nusselt number and the Reynolds number were deduced, one was for the case of air entering the tunnel duct laterally from one side and the other was for the case of air entering the duct from two sides. 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

The heat transfer from a flat plate situated at different locations in the wind tunnel channel has been

studied recently by Shalaby [1 and 2] and presented some formulas for the average Nusselt number. The flat plate was located vertically, as a single plate and as a plate between plates array, in the direction of air flow stream. In the case of the single plate 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:

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

where $4,300 \leq Re \leq 67,000$

In the case of the plate which is situated vertically between plates array, the obtained correlation was in the form:

$$Nu = 0.573 X^{0.29} Re^{0.53} X^{-0.055} \quad (2)$$

where X is the dimensionless distance from the right hand side wall of the wind tunnel and equals $[2x/b]$. Its value ranged between 0.2 and 1 at $6,680 \leq Re \leq 107,170$. It is observed that the dimensionless distance X appears in the obtained correlation, i.e. the Nu values depend on the test plate location (X).

This paper introduces the results of a subsequent investigation conducted to establish a correlation for the average Nusselt number of the hot plate situated, in the wind tunnel channel, at different locations between number of unheated plates array. In this work the air flow stream enters the wind channel from the lateral direction.

Recent advances in semiconductor technology prove that, as electronic components are made smaller, the quantity of heat that must be dissipated per unit volume of a device increases dramatically. The peak heat fluxes in a recent computer circuit are 10 times greater than those in an older computer [3]. Forced convection cooling of the circuitry is commonly used to maintain the desired operating temperature in such a device. This work is a part of a continuing program to understand better the limits of air cooling, and to develop methods for extending its usable range. Several investigators have attempted to increase the rate of heat transfer in a circuit board [4-10]. Shalaby, et. al, [11-13] 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 fundamental heat transfer literature on electronic equipment cooling [14 and 15] shows that the best location and the optimal spacing have not been determined for packages that are cooled by forced convection. The present work concerned with this problem, in the case of a lateral

air flow stream. This may develop 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 52.5 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. The air coolant has been drawn from the laboratory room and the air intakes are perpendicular to the wind tunnel axis as shown in Fig.(1).

EXPERIMENTAL TEST RIG AND PROCEDURE

Fig.(1) shows the test rig constructed for the planned experiments. The test rig consisted of a low-turbulance wind channel, in which air from the laboratory room is drawn through the system by a downstream blower (10). The test section is existed within the channel a 90 cm from the channel entrance. A throttle valve (13) is used to control the rate of air flow. In between the main channel (23) and the blower a flexible textile plastic connecton (9) is fitted in order to eliminate any vibration transmission to the honey comb section (8). The velocity of air drawn through the main channel is measured with the help of Pitot tube (5) and an inclined alcohol manometer (21) at the centers of nine imaginary equal areas into which the Pitot tupe is situated 50 cm upstream. The air velocity is, also, measured by the hot-wire probe (6) located 75 cm upstream. The difference in velocity values measured by the two methods is about ± 1.0 . Fig.(1-a) shows the layout of the constructed test rig. The heated test plate (7) is situated vertically between an array of unheated wooden slapes. This plates are kept vertically by the help of wooden slapes (25) as shown in Fig. (1-a). The leading edges of the wooden slapes (25) have a triangular shape with 15 deg, on the horizontal surface to avoid the air disturbance motion near the test plates entrantces.

The array unheated plates have the same dimensions as thoes of the a life one. The a life plate takes places far from the right vertical wind channel wall equal to $x=12.5, 22.5, 32.5, 42.5,$ and $62,$ mm. The distance from the centers of each adjucent plates 10 mm.

The test plate as shown in fig.(2) is made from two polished aluminum sheets (1), each of them has 1.0 mm thickness. Nine thermocouples copper-constantan (5) are brazed fixed on the back surface of each plate. The back surface is divided into nine imaginary equal roctangular areas. The thermocouples are made from 30 gauge

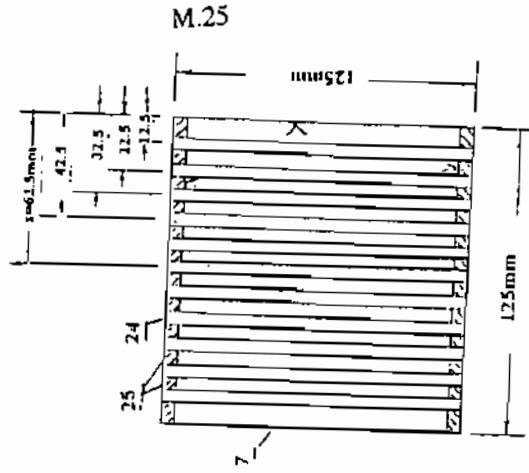


Fig. (1 - a) Section at B - B

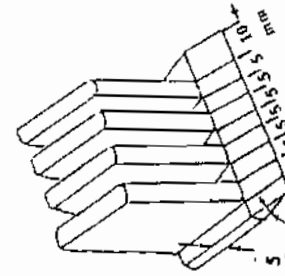


Fig. (1 - b) Construction of Test Section

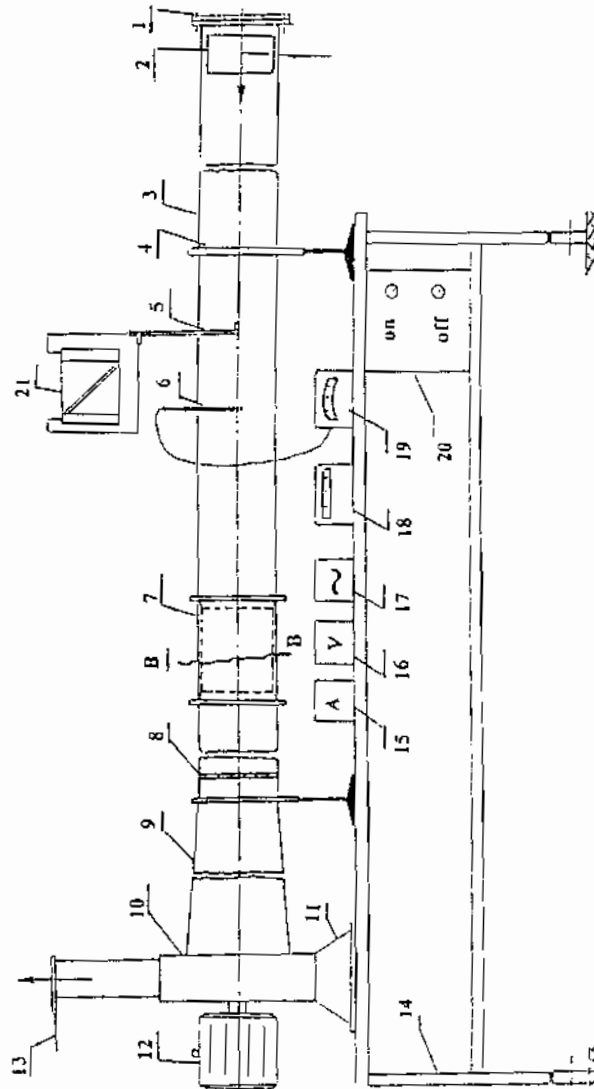


Fig. (1) Experimental Test Rig

- | | |
|------------------------------------|--------------------------------------|
| 1- Closed wooden plate | 3- Straight channel of 900 mm length |
| 4- Carriers | 6- Hot wire anemometer probe |
| 7- Tested plate | 9- Cloth connection part |
| 10- Down stream blower | 12- Electric motor |
| 13- Control discharge gate | 15- Ammeter |
| 16- Voltmeter | 18- Temperature record |
| 19- Hot wire anemometer | 21- Inclined manometer |
| 22- Copier constantan thermocouple | 24- Wooden plate |
| | 25- Separation wooden slaps |

wires.They fixed in thin slots (0.5 mm deep)cut on the under side of the plate the htermocouples calibration was performed .the average of the nine local temperatures is obtained for each face of test section plate as follows:

$$T_W = \sum_{n=1}^9 T_n / 9 \quad (3)$$

Where -- n -- is the location number of the measured temperature point

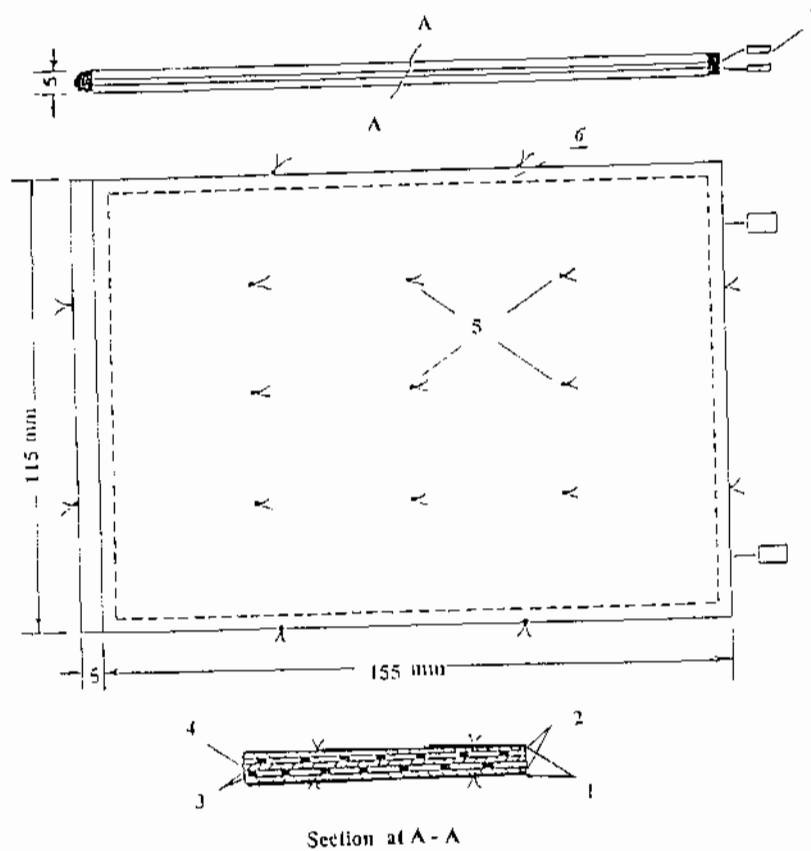


Fig. (2) Hot Plate Construction

- | | |
|----------------------------------------------|----------------------------------------|
| 1- Two aluminum plates of 1 mm thickness | 2- Two mica plates of 0.5 mm thickness |
| 3- Rectangular cross section electric heater | 4- Mica plate of 0.5 mm thickness |
| 5- Nine constantan thermocouple | 6- Glass wool insulation |
| 7- Electric heater connection | |

Since the local temperatures are sensed at a depth of about 0.5 mm from the top polished surface, they can be considered as aluminum outer surface temperature.

It is found that the difference between the calculated outer surface temperature and the measured one was less than 0.03 °C.

An electrical heater made of nickel-chromium heating wire is used for heat the test plate. The heating wire is wound around a threaded mica sheet (4). This sheet is sandwiched between two mica sheets (2). Each mica sheet has 153 mm length, 113 mm width and 0.5 mm thickness. The outside dimensions of the test plate are 155 mm length, 115 mm width and 5 mm thickness. The leading edge of the plate is rounded in the direction of flow to avoid the disturbance in the trailing edge. The test plate sides are insulated by three layers of glass wool tape. Eight thermocouples are placed and embedded at the mid-height of the surrounding tape (6). These thermocouples help in finding the conduction heat loss (Q_{cond}) from the plate sides. On other side, the mean bulk air temperature are measured by using two movable thermocouples facing each side. All the thermocouples are connected to a six-point-temperature recorder [(18)-Fig.(1)]. The channel walls temperatures are measured by three thermocouples existed on the right, left, upper walls of the wind channel, and the dry bulb temperature at the intake and the outlet of the working section are, also, measured.

An auto-transformer (17) is used to control the heat input as well as one voltmeter (16) and ammeter (15) as shown in Fig.(1).

During the course of this experimental work nearly 2.5 hours are needed to reach the steady state condition. the satisfaction of temperature reading is considered when there is no any change within a time period of about 15 minutes. The principle of conservation of energy is applied to give:

$$Q_{in} = Q_{cond} + Q_{conv} + Q_{rad} \quad (4)$$

The amount of heat conducted away through the thermocouples wires and the power leads can be neglected because they have very small cross sections and the various connections in the apparatus are made of insulated materials.

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 [16,17], 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;

$$h = (Q_{in} - Q_{cond} - Q_{rad}) / \{ A(T_w - T_a) \} \quad (5)$$

The probable error in finding the average heat transfer coefficient was estimated to be about $\pm 10\%$. 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 . So, the Reynolds number values have been corrected to meet this blockage effect in the air flow cross-sectional area.

RESULTS AND DISCUSSION

For finding the average heat transfer coefficients and their correlation with air flows, 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 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 (5). 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 changed 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 6,806 and 108,837 and the values of x chosen are 12.5, 22.5, 32.5, 42.5 and 62.5 mm.

The results of different plate locations are displayed in the form of Nusselt number versus Reynolds number, as shown in Fig.(3), in the case of air flows laterally into the wind duct from one side . The same results have been reported also, in the case of air flows laterally into the wind channel from two sides ,as shown in Fig.(4). Each figure is plotted for values of x equal to 12.5, 22.5, 32.5, 42.5 and 62.5 mm. The figures, also, have one dashed line obtained from correlation (1) as well as one solid line obtained from correlation (2) and the dotted line represents the following correlation[17];

$$Nu = 0.6795 Pr^{0.33} Re^{0.5}$$

(6)

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

The above 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 (1), the dotted line of equation (2) and the dashed line of equation (6). This may be because of the lateral air flow effect. It is also seen that the slope of the present results line is higher than the slopes of the three lines obtained by equations (1), (2) and (6). The present data show some approach with the same obtained by equation (1) at low Re values and at high Re values the Nu values are, in general, higher than the same obtained by equation (1). The recorded data show a remarkable increase in Nu values ranged between 5% and 20%. 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], as well as the effect of the lateral air flow into the duct.

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 the same Figure. The main observation, coming out for the case of $x = 12.5$ mm is valid for the cases of x equals 22.5, 32.5, 42.5 and 62.5 mm. One may observe that the slope 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. The results reported in Fig.(4) are for the case of air flow laterally from two sides of the tunnel duct. These results show the same main observations like those reported in Fig.(3) for all x 's values. The new only observation shown in Fig.(4) is the Nu data obtained in Fig.(4) higher than the same obtained in Fig.(3) by values ranged between 2% and 6%.

In all, 154 data points obtained are represented in Fig.(5) and 147 data points obtained are represented in Fig.(6). One may observe that, the Nu values, generally, increases with Re. The data show a weak dependence of Nu

on the test plate location (x). Along the course of this work the heat input to the test plate heater varies between 3.70 W and 240.0 W.

Figure (7) shows Nu values versus x at Re equals 100,000, 80,000, 60,000, 40,000, 20,000 and 1,000. One may observe that the Nu values show a very little increase 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 that comes from [18].

General Heat Transfer Correlation:

The results of the plate locations 12.5 mm to 62.5 mm may be taken as one group as shown in Fig. (5 and 6). A trial was made to find out a correlation between Nu and Re. The obtained correlation was in the form:

$$Nu = C Re^m \quad (7)$$

where $6,806 \leq Re \leq 108,837$ and $C = 1.35$ and $m = 0.49$, in the case of air flow into the tunnel duct from one side, and $C = 0.85$ and $m = 0.53$, in the case of air flow into the tunnel duct from two sides.

One may observe that the dimensionless distance X (x/b) does not appear in the obtained correlation, i.e. the Nu values independent of the test plate location (X), as previously observed. The experimental data, shown in Fig. (5 and 6), agree with the values obtained from correlation (7) by $\pm 15\%$.

CONCLUSIONS

As a result of the present investigation the following conclusions have been derived:

- 1- As Reynolds number increases the average Nusselt number value increases, i.e., the average heat transfer coefficient increases with Reynolds number.
- 2- There is no significant effect on the average heat transfer coefficient when the air enters the channel laterally from one side or from two sides.
- 3- The air flow into the tunnel duct from the lateral sides has a significant effect on the average heat transfer coefficient, specially at high Reynolds number values.
- 4- The test plate location does not affect much the average heat transfer coefficient.

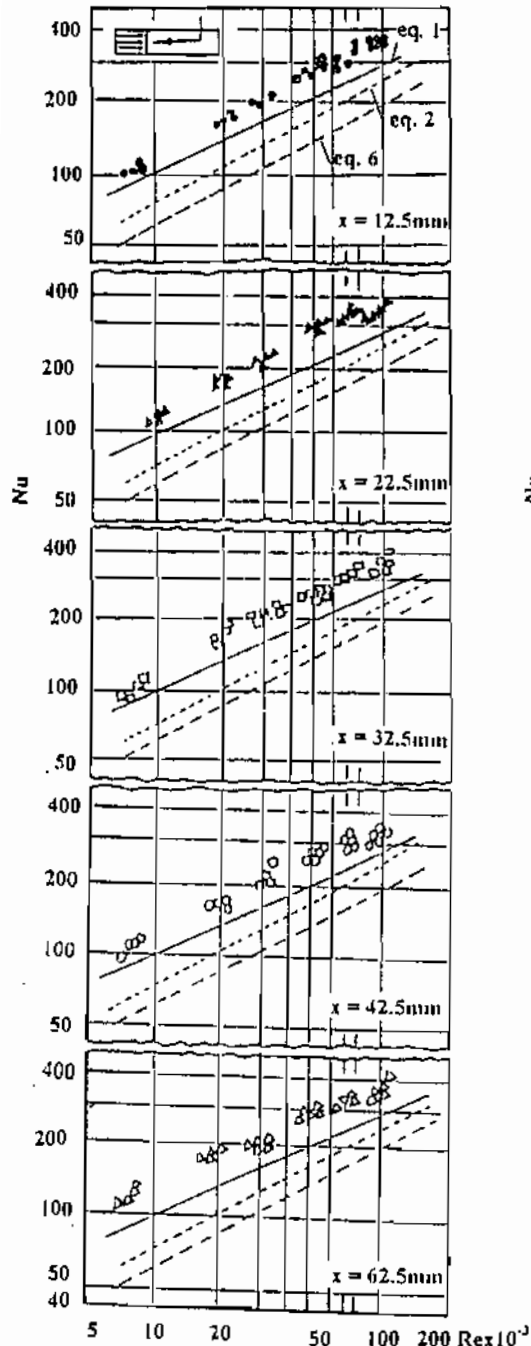


Fig (3) . Relation between Nusselt number versus Reynolds number for a heated plate at different locations with one air intake window .

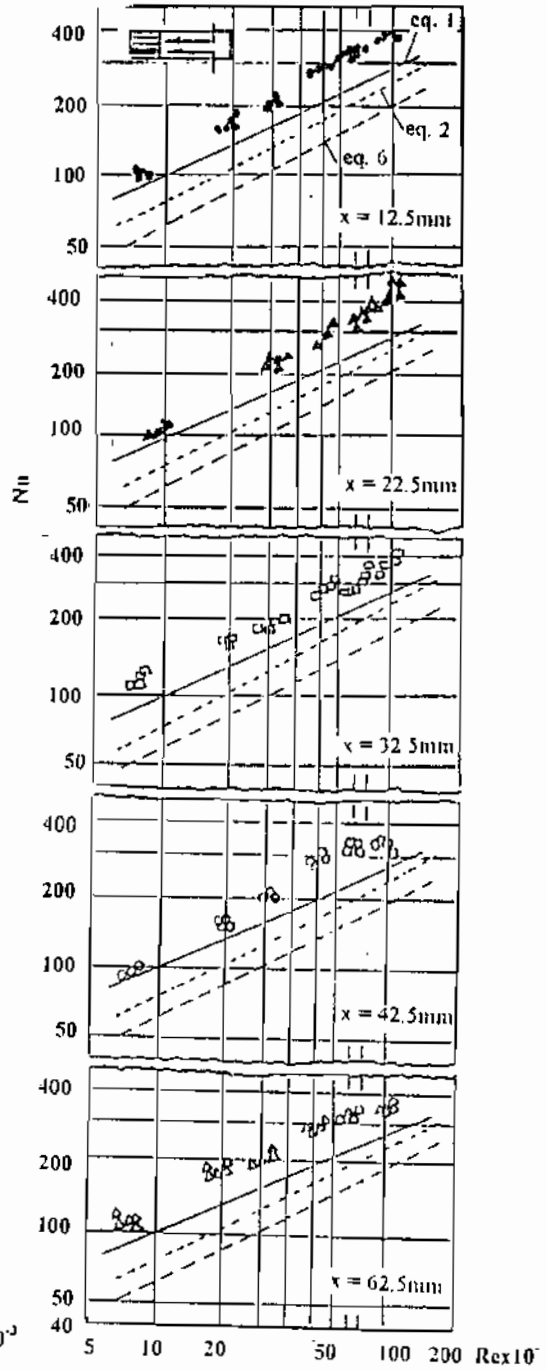


Fig (4) . Relation between Nusselt number versus Reynolds number for a heated plate at different locations with two air intake windows .

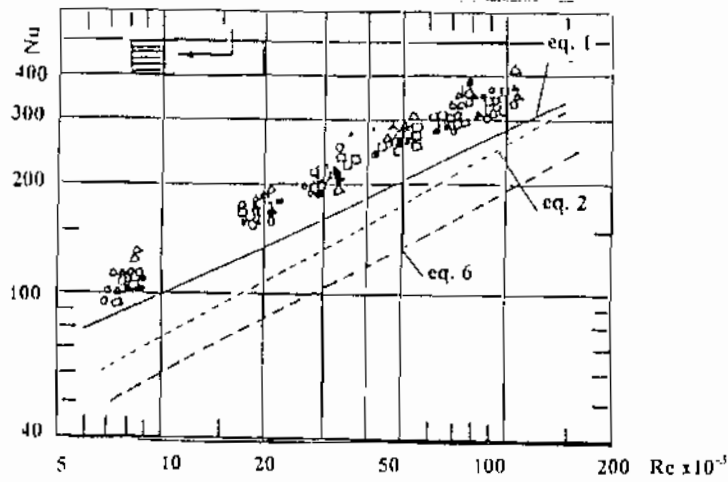


Fig (5) . Average Nusselt number versus Reynolds number at different locations with one air intake window .

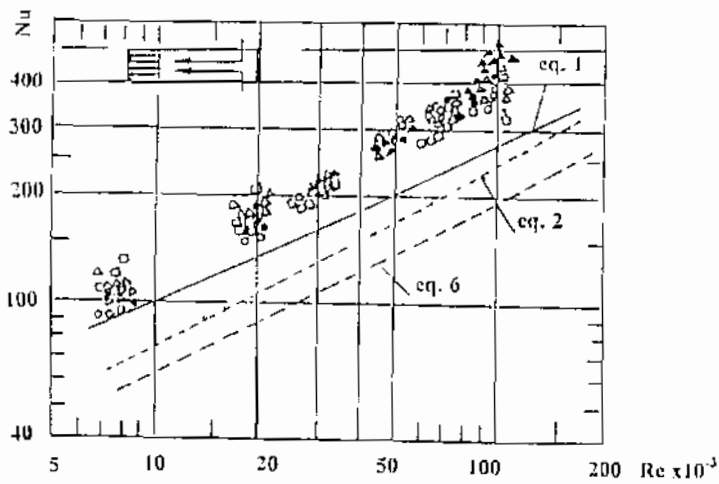


Fig (6) . Average Nusselt number versus Reynolds number at different locations with two air intake windows .

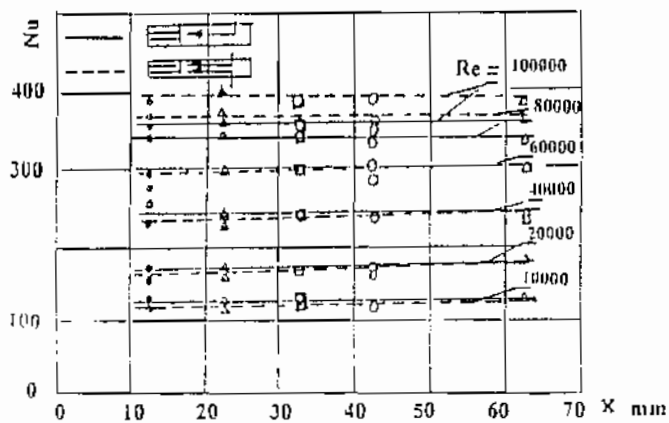


Fig (7) . Relation between Nusselt number and plate locations x, in mm at different values of Reynolds number

NOMENCLATURE

The symbols used in the paper have the following meanings:

A = Plate surface area, $[m^2]$,
 b = Wind tunnel channel width, $[m]$,
 C_p = Specific heat of the fluid, $[kJ/kg.K]$
 h = Average heat transfer coefficient, $[W/m^2.K]$,
 k = Thermal conductivity of the fluid, $[W/m.K]$,
 L = Test plate length, $[m]$,
 Nu = Average Nusselt number, $[h.L/k]$,
 Pr = Prandtl number, $[\mu C_p/k]$,
 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_a = 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, $[m^2/s]$,

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