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HEAT TRANSFER AND FLOW VISULIZATION OF SEPARATED REATTACHED AIR FLOW OVER REVERSED RECTANGULAR FLAT PLATE

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خلاصة

يتضمن هذا البحث الدراسة العمليه لانتقال الحرارة بالحمل القسري من لوح مستطيل مستو ساخن مصنوع من صفيحة من الالمنيوم المصقول الى تيار هواء مضطرب. وهذا اللوح معلق فى نفق هوائى بحيث تكون زاوية التعاكس بين مقدمة اللوح واتجاه سريان الهواء معكوسه وتتغير من صفر الى ٦٠ درجة . واثناء الدراسة تم تغيير رقم رينولد من ٧٥٠٠٠ الى ٢٢٥٠٠٠ . وقد اوضحت الدراسة ان متوسط معامل انتقال الحرارة بالحمل دالة فى رقم رينولد وزاويه التعاكس للسطح المستو . وانه بزيادة زاويه التعاكس تتناقص قيمة معامل الانتقال الحرارى بالحمل عامة ماعدا بعض الحالات الخاصة عند زوايا تعاكس ١٠ر٥٠ درجة . وقد عملت دراسة لتصوير خطوط سريان تيار الهواء على سطح اللوح وبيان مناطق انفصال واتصال التيار الهوائى على السطح . وقد صيغت النتائج فى معادلة تجريبيه لامكانية حساب معامل انتقال الحرارة المتوسط بالحمل.

ABSTRACT

Average heat transfer coefficients during forced convection air flow over reversed-rectangular flat plate have been experimentally determined. The experiments are carried out for a constant surface temperature and covered reverse angles from 0 to 60 deg, and Reyonolds numbers from 7.5×10^4 to 22.5×10^4. The results show that the average heat transfer coefficient is a function of Reyonlds number and the reveres angles. Correlation equation for various angles of reverse is suggested.

INTRODUCTION

Forced convection heat transfer to airflow over an arbitrarily oriented plate represents both a three-dimensional boundary-layer research problem and a prototype situation for numerous applications. The applications include many of contemporary interest such as wind-related heat losses from solar collector plates and from walls and roofs of buildings. For these applications, the quantity that is of mostdirect relevance is the average heat-transfer coefficient. Information on average heat-transfer coefficients for airflow over square plates is determined experimentally in [1]. It is found there that the heat transfer coefficients are quite insensitive to the orientation of the plate relative to the airstream. Furthermore, it is domenstrated that the heat transfer coefficients that have been conventionally employed for the calculation of wind-related heat losses from flat plate solar collectors are seriously in error.

Recently, Motwani et al. [2] have studied experimentally the same problem using rectangular plates. Tripping wires are used at the edges to ensure that a turbulent boundary layer prevailed over the plates. Their experiments are carried out for a constant surface temperature and covered two plates of aspect ratios equal to 2/3 and 3/2 for Re* ranging from 2×10^4 to 3.5×10^5. They concluded that the average heat transfer coefficient is insensitive to the aspect ratio and angle of yaw.

Abdel-Salam et al.[3] studied the forced convection heat transfer from a rectangular flat plate to an air stream. In their work the attack angle is ranging between 0 deg and 70 deg and the Reynolds number (Re) is varying from about 78, 500 to 230,000 . Based on their extensive heat transfer experiments on a rectangular plate held in a wind tunnel , they suggested some useful correlations.

More recently, Shalaby et al. (4,5) have studied experimentally the average heat transfer coefficients over an inclined and yawed rectangular plate to an oncoming air stream. The experiments covered angles of attack from 0 to 45 deg, angles of yaw from 0 to 45 deg and Reynolds numbers (Re_L^*) from 68,000 to 220,000 and carried out for a constant surface temperature. Wooden sharp edge at the plate front is used to decrease the bluntness effect over the plate. Tripping wires are also used at the edges to ensure that a turbulent boundary layer prevailed over the plate. In their work they have suggested two useful correlations.

The purpose of the present study is to investigate experimentally the heat transfer characteristics in the separated, reattached, and redeveloped regions of three-dimensional incompressible air flow around a reversed rectangular flat plate with wooden sharp nose. The flat plate is held in a wind tunnel channel and inclined and does not, in general, span the channel so that transverse flow may occur. The reverse angle is ranging between 0 deg and 60 deg. The experiments for determining average heat transfer coefficient have been conducted for constant surface temperatures over Reynolds numbers (Re_L^*) ranging from about 7.5×10^4 to 22.5×10^4. The flat plate is made from aluminum of 150 mm length x 120 mm width x 2 mm thickness. The objective of the present work is to provide a correlation or correlations for a three-dimensional boundary layer on the reversed plate.

EXPERIMENTAL APPARATUS AND PROCEDURE

The wind tunnel used in the present study is the same as that employed in the previous work by the authors [4 and 5]. The test plate (2 mm thick, 120 mm wide, and 150 mm long) is made of a polished aluminium plate (see Fig. 1). The aluminium plate back surface area is divided to nine imaginary equal rectangular areas, each of 50 mm length and 40 mm width. In the center of each area a copper-constantan thermocouple, made from 30 gauge wires, is fixed in thin slots (1 mm deep) cut on the underside of the plate. Thus the average of nine local temperatures is obtained. The temperatures are sensed at a depth of about 1 mm from the top polished surface. As such these values can be taken to represent the top surface temperatures because of the use of aluminium test plate in which the difference between the estimated top surface temperatures and the measured values are found to be in general less than $0.025^{\circ}C$. The plate is heated electrically by means of the main heater (10). This heater consisted of a nickel chromium heating wire wound around between two sheets of mica (9) each of 150 mm long x 120 mm wide x 1.5 mm thick. Measurements in the main series of tests show that the variation in plate surface temperature is almost always less than 2.5 % of the difference between the average plate temperature and the free-stream temperature The use of an aluminium test plate coupled with a closely wound heater ensured that the desired condition of a constant surface temperature is obtained.

To prevent the heat loss from themain heater back, a guard heater (4) is used. The combination of guard heater is mostly the same as the one used for the main heater and is shown in Fig.1. The heat input to each of the main and guard heater is controlled by using two auto-transformers as well as two voltmeters and two ammeters. A bakelite plate (7) of dimensions 150 mm long x 120 mm wide x 16 mm thick is sandwitched between two aluminium sheets each of dimensions 150 mm long x 120 mm wide x 2 mm thick, and the whole set is placed between the set of the main and the guard heaters as shown in Fig. 1. For a fixed main heater input, the guard heater input is regulated so as to maintain as small a temperature difference as possible less than 0.2 ^{o}C across the bakelite plate thereby ensuring that the heat flow from the plate bottom is negligible. Eight-junction thermocouples used for this pupose had four junctions on each side of the bakelite plate. The junctions are located at the centers of the four imaginary equal rectangular areas into which the plate sides are divided. These thermocouples are conneted to digital multimeter with an accuracy of 0.001 mV. Bakelite strips (16 mm thick) are fixed on all the four edges of the test plate (see Fig. 1). A wooden sharp edge (1) is fixed at the plate leading edge. The overall dimensions of the assembly are 237 mm x 152 mm x 45 mm thick. Eight thermocouples are placed and embedded at 10 mm depth in the bakelite frame and 10 mm far from its outer surface. On the outer surface of the bakelite frame eight thermocouples are placed and embedded at the mid-height of the frame surface (see Fig. 1). The purpose of these thermocouples is to determine the heat loss by conduction (q_s) from the heating plate sides.

The air stream temperatures before and after the test plate, and the wind tunnel surface temperature are also measured by set of thermocouples (1) located as shown in Fig. 2. The test plate assembly (2) is supported on an adjustable wooden bar (3) that extends in the wind tunnel (4) and supproted by vertical sides. The wooden bar is adjusted to fix the reverse angle of the plate at 0, 5, 10, 20, 30, 40, 50 and 60 deg , as shown by the pointer (5) on the protractor (6).

In order to estimate the heat lost by radiation (q_r) the average value of emissivity of 0.24 for polished aluminium plate as taken from [2], in which they reported that no significant dependence of emissivity on temperature is observed. The summation of estimated values of q_s and those of q_r are of order 20 to 40% of the input power to the main heater neglecting the heat loss from the back (q_b) of the test plate. A steady state is usually achieved after about three hours. The average convective heat transfer coefficient is determined from the expression

$$h = \frac{q_{in} - q_s - q_r - q_b}{A\,(T_s - T_\infty)} \qquad \ldots\ldots(1)$$

The probable error in finding the average heat transfer coefficient is estimated to be about 9%. The projected areas of the test plate on a vertical plane perpendicular to the tunnel axis are calculated by the following correlation

$$A_p = 346.5 \sin\gamma + 67.5 \cos\gamma \qquad \ldots\ldots(2)$$

It is found that the maximum blockage of the wind tunnel free stream cross

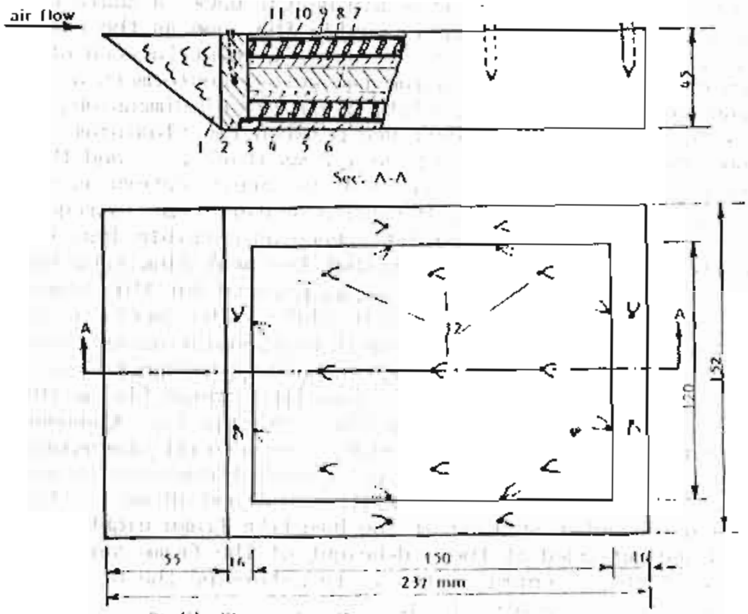

Fig.(1) . Plate test section

1- sharp edge, 2- insulator box, 3- aluminium bottom, 4- electric guard heater, 5 and 9- plates of mica, 6 and 8- aluminium plates, 7- Bakelite plate, 10- electric main heater, 11- tested plate, 12- copper-constantan thermocouples.

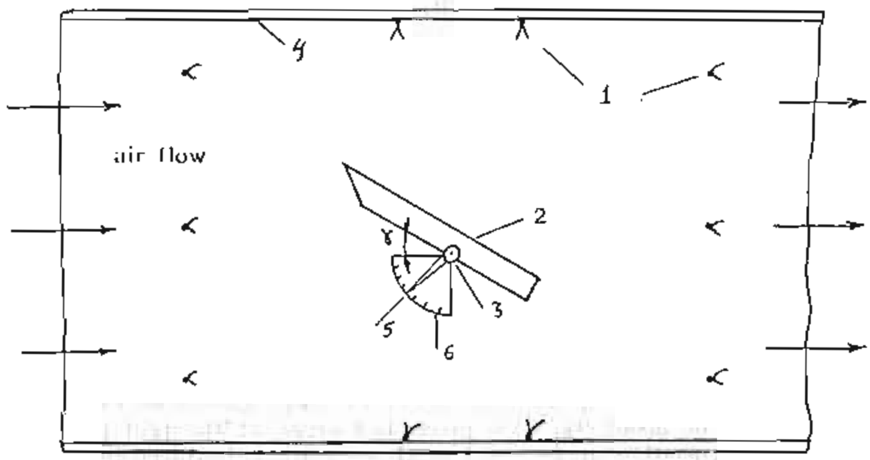

Fig.(2). Definition sketch for a test plate placed in the wind tunnel channel.

1- Thermocouple, 2- test plate assembly, 3- wooden bar, 4- wind tunnel channel, 5- pointer, and 6- protractor.

section area at higher values of γ is about 42.9%. Test and Lessmann[6] have reported that their heat transfer results with and without blockage differ by a maximum of 7%. Therefore, the blockage does not affect the heat transfer results to some extent.

The used open circuit wind tunnel has a cross section of 30-cm-square, in which air from the laboratory room is drawn through the system, by a downstream blower. The flow rate is controlled by a throttle valve. The velocity of the air stream drawn through the system is sensed by a hot-wire probe located 75 cm up-stream.

RESULTS AND DISCUSSION

For the determination of the average heat transfer coefficients and their correlation with airflow over the reversed rectangular flat plate, some quantities are measured for each data run. The power input to the main heater, the heat lost by conduction from the test plate sides, the average plate surface temperature, the airflow stream velocity and the free stream temperature are recorded. The heat lost due to unbalance between the main and guard heaters is neglected (about 0.003%) because the temperature difference on the sides of the bakelite plate placed between the two heaters is kept very small (less than 0.2 ^{o}C). Net rate of heat transfered by convection is used to calculate the average heat transfer coefficient from equation (1). Since, the flow is a three-dimensional flow, the characteristic dimension used in the calculation of both Nusselt and Reynolds numbers is L^* defined by $L^* = 4A/C$.

During the course of the experimental work, Reynolds number is varied from 7.5×10^4 to 22.5×10^4, the reverse angle from 0 deg to 60 deg and the main stream velocity from 11.2 to 25.3 m/sec. The temperature difference between the test plate surface and the oncoming air ΔT is varied from 15 to 46 ^{o}C at a given Reynolds number. In all, 57 data points are obtained and in addition, a number of experiments are repeated.

The results obtained in terms of Nusselt number (Nu_L^*) versus Reynolds number (Re_L^*) for reverse angles of 0, 5, 10, 20, 30, 40, 50, and 60 deg are shown in Fig. (3). Inspection of the figure shows that, the Nusselt number value increases in general with Reynolds number. The heighest Nusselt number values are displayed for $\gamma = 0$ deg, and below these values by about 12% the results of the reverse angle equal to 10 deg are displayed. The results obtained at $\gamma = 5$ deg lie below the data of $\gamma = 10$ deg by about 8%. As known the situation of the air flow on the test plate is a three-dimensional turbulent boundary layer flow. Therefore a significant portion of the air flow escapes over the lateral edges due to the limited size of the test plate. On the other hand, due to the blunt side edges of the test plate, the flow jumps up at the oncoming air plate sides and reattaches the plate after certain distances, i.e. flow separation occurs . Hence, as the reverse angle (γ) increases from 0 deg to 5 deg the leading edge blunt increases and the air escaping from the lateral edges still has a considerable amount. As the reverse angle increases to 10 deg the portion of the air flow escaping from the lateral edges decreases. Because the oncoming flow from the lateral edges brakes down the portion of the airescaping. Therefore the heat transfer results displayed at $\gamma = 10$ deg lie above the same obtained at $\gamma = 5$ deg. More inspection of the figure shows that, as the reverse angle increases the heat transfer coefficient decreases. The results obtained at γ equal to 20 deg and 40 deg are more or less the same as the figure shows.

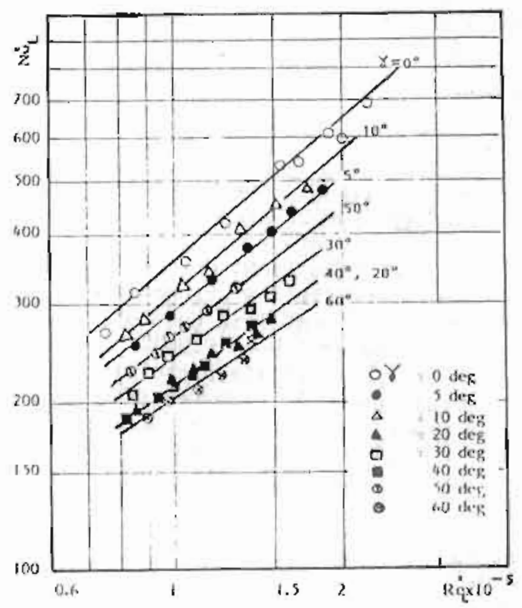

Fig. (3). Variation of Nusselt number with Reynolds number.

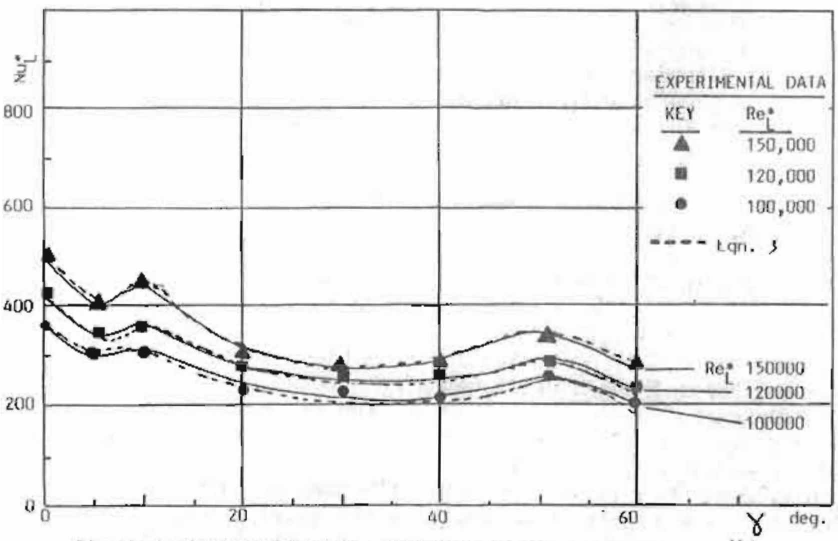

Fig.(4) Variation of Nusselt number Nu_L^* with the reversed angle (γ).

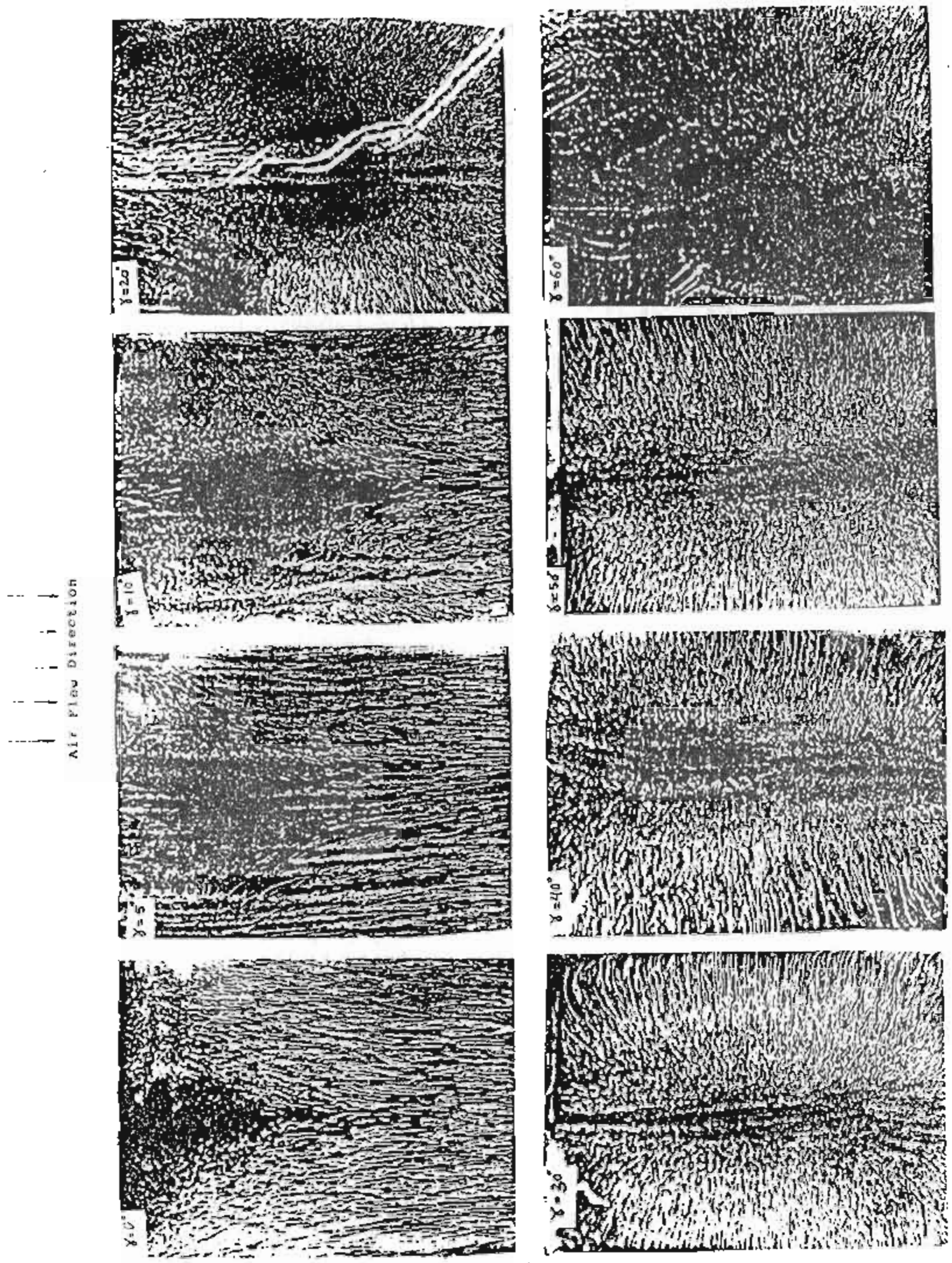

Fig.5. Fluid Flow Patterns for Reverse Angles γ of 0°, 5°, 10°, 20°, 30°, 40°, 50° and 60°, and for $Re_L = 11.5 \times 10^4$.

One may also observe that the displayed data at γ = 50 deg lie below the data obtained at γ = 5 deg, leaving the expected location below the results obtained at γ = 40 deg. This may be due to the boundary layer separation in which the pressure gradient adverses on the test plate surface and hence, the transverse flow increases and affects the stagnation zone appears along the test plate, as shown in Fig. (5), and causes the increase in the average heat transfer coefficient at the reverse angle equal to 50 deg.

Figure (4) shows the variation of the Nusselt number values with the reverse angles of the test plate for Reynolds number values equal to 10^5, 12×10^4 and 15×10^4. It is observed that the values of Nu_L^* increases at two locations; one at the reverse angle equal to 10 deg and the other location at γ = 50 deg, as discussed above. Figure also shows that the Nusselt number values increase with Reynolds number.

FLUID FLOW PATTERNS

The patterns of fluid flow adjacent to the plate surface are made visible by employing the oil/lampblack technique. Under the action of the shear stresses exerted by the air on the surface, an initially uniform film of a mixture of oil and lampblack is moved along the surface in the flow direction. The result is a pattern of streaks which reveal the flow path of the air as it passes over the surface. In a stagnation region, the oil and lampblack mixture remains as it was initially applied, so that a black region without streaks is indicative of stagnation conditions.

The flow visualization studies are carried out for Reynolds number 11.5×10^4, and the plate orientations encompassed reverse angles γ of 0^0, 5^0, 10^0, 20^0, 30^0, 40^0, 50^0 and 60^0. Attension will first be turned to the results for zero reverse angle as presented in Fig. 5. When the plate is situated parallel to the flow (γ = 0 deg), there is a stagnation zone in the entrance region of the plate with a surrounding region of longitudinal inflow. As the plate reverse angle (γ) increases to be 5 deg, the stagnation zone becomes wider and progressively moves forward. At the reverse angle equal to 10 deg, the stagnation zone has a somewhat elliptical shape and becomes in general narrawer. One may also observe that, the streaks pattern shows some lateral inflow. As the plate reverse angle increases to be 20 deg, the stagnation zone progressively moves forward and is concentrated along the half right and lift-hand edges of the plate and the zone extended to cover most of the plate core area. The pattern of the streaks shows that, most of the air comes from the lateral edges and the back one, i.e. the incoming air is mostly a transverse flow, but it seems a weak flow. In case, the plate has the reverse angle 40 deg the stagnation zone takes the form of a band along middle portion of the plate. On the other hand the pattern of streaks indicate that the inflow extended along the right and left hand edges. However, in case of the reverse angle equal to 30 deg, the streaks pattern becomes more stronger and shows that the two sides inflow have more effect and the stagnation zone occupies a narrow band along the plate. So, the heat transfer coefficient, in this case, will be higher than the above two cases. As the reverse angle of the plate increase to be 50 deg, the transverse flow increased and affected the stagnation zone and appeared along the test plate as shown in the figure in the form of a longitudenal band narrow than in the case of 40° but more pronounced than in the case of 30°. Finally, the figure shows that, at the reverse angle equal to 60 deg, the stagnation zone extended to cover most of the test plate surface, and the pattern of streaks indicate that, there is a somewhat weak transverse flow at the plate corners.

From overall inspection of the two Figs. (3 and 5), one may observe that as the Reynolds number increases the Nusselt number values ingeneral increase. It is also seen that the stagntion zone area increases ingeneral with the reverse angle, i.e. the air flow attachement with the test plate surface decreases and concequently the heat transfer coefficient in the separation zone decreases. As such the average heat transfer values of the plate decreases. One may also observe that, whenever the stagnation zone exists on the plate the boundary layer separation takes place due to the pressure gradient adverses and the transverse flow increases from the lateral edges as well as the back one as shown in Fig. (5). It is also observed that the average Nusselt number value ehanges with the change of the stagnation zone area, in which it increases with the decrease of the separation zone area and vice verse. This appeares in the case of the data obtained at $\gamma = 5°$, in which the displayed values are lower than the same obtained at 0° and 10°. While the reported data for the two cases of $\gamma = 20°$ and 40° are more or less the same and both of them lie below the values obtained at $\gamma = 30°$. However the results of $\gamma = 50°$ are higher than the displayed values at $\gamma = 0°$, 5° and 10° as shown in Fig. (3). On the other hand, the values of 60° lie below the whole set of data obtained and it also shows the largest stagnation zone area in Fig (5).

CORRELATION.

An attempt is finally made to correlate the data obtained in the present study. Such a correlation is quite useful from a designer's standpoint. The average Nusselt number (Nu*) is correlated with the other relevant govering parameters of the test plate, namely Reynolds number (Re_L^*) and reverse angle (γ).

The following correlation is obtained

$$Nu_L^* = 0.07\,[(1+\gamma')^{-1.25}(\cos\gamma)^{-0.61} + 0.1\cos 105\sqrt{\gamma}\,]\,Re_{L^*}^{0.74} \qquad \ldots(3)$$

$$7.5 \times 10^4 \leqslant Re_{L^*} \leqslant 22.5 \times 10^4$$

$$0 \text{ deg} \leqslant \gamma \leqslant 60 \text{ deg}$$

where γ' in readians. The above equation predicts the values of the data within $\pm$ 5%.

In equation (3), the exponent of 0.74 on Re_L^* clearly suggests the presence of a tubulent boundary layer. The term $(1+\gamma')(\cos\gamma)$ may be attributed to the increasing size of the separation bubbles and the decreased lateral outflow as γ increases.

CONCLUSIONS

The research described here constitutes a comprehensive study of heat transfer and flow visulization of separated reattached air flow over reversed rectangular flat plate. The plate has reversed angle (γ) ranging between 0^o and 60^o. The Reynolds number (Re_L^*) based on the hydrodynamic characteritic length (L^*) was varied from 75,000 to 225,000.

Out of this study, one can conclude that the average heat transfer coefficient value increases ingeneral with Reynolds number and on the other hand it decreases with the reverse angle. As the revese angle increases the stagnation zone area on the plate surface increases (poor heat transfer area) and consequantly the transverse flow from the lateral edges as well as the back edge increase due to the effect of the generated adverse pressure gradient. One may also conclude that the effect of the transverse flow sometimes is strong enough to overcome the separation zone as it happens with the reverse angle γ = 50 deg, as such the average heat transfer coefficient shows a markeable increase.

NOMENCLATURE

A	plate area, m^2
A_p	projected area of the test plate, Cm^2
C	circumference of the plate, m
h	average heat transfer coefficient, W/m^2 K
k	thermal conductivity, W/m K
L	plate length, m
L^*	characteristic length, 4A/C, m
Nu_L	Nusselt number, hL^*/k
q_b	heat loss from the plate bottom, W
q_{in}	power input to test plate, W
q_r	heat loss by radiation from the test plate, W
q_s	heat loss from the plate sides, W
Re_L^*	Reynolds number, $U.L^*/\nu$
T_s	average plate surface temperature, ^oC
T_∞	free-stream temperature, ^oC.
ΔT	temperature difference, $(T_s - T_\infty)$, ^oC
U_∞	local average free stream velocity, m/s.
γ'	reverse angle, in radians.
γ	reverse angle, deg
ν	kinematic viscosity, m^2/s
ρ	density, kg/m3.

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