Mansoura Bulletin Vol. 11, No. 2, December 1986

TRIPPING WIRES EFFECT ON HEAT TRANSFER DURING WIND FLOW OVER RECTANGULAR INCLINED AND YAWED FLAT PLATE

By

SHALABY, M.A. and ARAID, F.F.
Mechanical Power Engineering Department,
Mansoura University, Egypt

ABSTRACT

Average heat transfer coefficients during forced convection air flow over inclined and yawed rectangular plate have been experimentaly determined. The study is concerned with the three dimensional behavior of a turbulent boundary layer on the heated flat plate. Tripping wires at the edges ensured that a turbulent boundary layer prevailed the plate. Wooden sharp edge at the plate front is also used to decrease the bluntness effect over the plate. The experiments covered angles of attack from O deg to 45 deg, angles of yaw from O deg to 45 deg, and Reynolds numbers from 64,000 to 220,000 and are carried out for a constant surface temperature.

The results obtained are compared with the available experimental data. The values of average Nusselt numbers are fitted in correlations. These correlations indicate that the Nusselt number is a significant function of the Reynolds number and the angle of attack and it is insignificant function of the angle of yaw.

INTRODUCTION

The present study is concerned with the three - dimensional behavior of a turbulent boundary layer on a flat inclined and yawed to an oncoming air stream at various angles of attack and yaw. Tripping wires at the edges are used to ensure that the turbulent boundary layer prevailed over the plate.

Very few investigators studied the three - dimensional flow over an inclined plate. Sparrow and his co - warkers [1,2] studied the problem and the motivation behind their work was the need for obtaining suitable correlation for the wind heat transfer coefficient on the cover of a solar flat plate collector. Based on their extensive mass transfer experiments on square and rectangular plates held in a wind tunnel, they suggested the following correlation

$$St = 0.86 \ Re^{-0.5} \ Pr^{1.5} \quad , \qquad (1)$$

$$25 \text{ deg} \leq \alpha \leq 90 \text{ deg} , \quad \text{and}$$

$$0 \text{ deg} \leq \psi \leq 45 \text{ deg}.$$

The exponent - 0.5 on Re in equation (1) suggests that the correlation is valid for a laminar boundary layer. One would not normally expect a laminar boundary layer in actual solar installations. One may also observe in Sparrow's experiments the reported data are , in general, for angles of attack more than 25 deg. The optimal slope of a collector would be much lower in the tropics compared to that in temperate zones. Thus it would be desirable to obtain data at smaller angles of attack as well.

Recently, Motwani et al. [3] have studied experimentally the average heat transfer coefficients during forced convection over a flat plate oriented to an oncoming air stream at various angles of attack and yaw. Their experiments were carried out for a constant surface temperature and covered two plates of aspect ratios equal to 2/3 and 3/2 for Reynolds (Re_{L*}) number ranging between 2×10^4 to 3.5×10^5. They have suggested the following dimensionless correlations:

$$Nu_{L*} = 0.056\,(1-\alpha/44.2)\,(Re_{L*})^{0.77};\ 0\ \text{deg} < \alpha < 15\ \text{deg} \tag{2}$$

$$Nu_{L*} = 0.084\,(Re_{L*})^{0.68};\ 30\ \text{deg} < \alpha < 45\ \text{deg} \tag{3}$$

In correlation (2), the exponent of 0.77 on Re_{L*} suggests the presence of a turbulent boundary layer at low angles of attack, due to the effect of the tripping wires fixed at the plate edges, and the term $(1-\alpha/44.2)$ may be attributed to the decreasing size of the separation bubble and the increased lateral outflow as (α) increases. While in correlation (3), the value of the exponent on Re_{L*} is reduced to 0.68. This indicates relaminarization of the boundary layer because of the favorable pressure gradient at higher angles of attack. They also concluded that the average heat transfer coefficient is insensitive to the aspect ratio and angle of yaw.

More recently, Shalaby et al. [4] have studied experimentally the average heat transfer coefficients over inclined and yawed rectangular plate to the air flow direction. The experiments covered angles of attack from 0 to 45 deg, angles of yaw from 0 to 45 deg, and Reynolds numbers from 68,000 to 220,000 and were carried out for a constant surface temperature.Wooden sharp edge at the plate front was used to decrease the bluntness effect over the plate. They have suggested the following dimensionless correlation

$$Nu_{L*} = 0.0022\ Re_{L*}^{1.022}\,(1+\alpha)^{0.02}\,(1+\Psi)^{0.73}, \tag{4}$$

$0\ \text{deg} \leq \alpha \leq 45\ \text{deg}$, and

$0\ \text{deg} \leq \Psi \leq 45\ \text{deg}.$

where α and Ψ are in radiant. In correlation the Nusselt number values are sensitive to the angle of yaw (Ψ) due to the presence of the wooden sharp edge at the plate front ,which reduces the bluntness effect of the plate.

Jurges [5] proposed a dimensional correlation of the form

$$h = a + b\,U_\infty^{n} \tag{5}$$

However, equation (5) suffers from two drawbacks. Both of them are for the case of parallel flow of air over a square vertical plate and as such do not reflect the effect of angle of attack or yaw on heat transfer. The second and more important point is that the equation is independent of the plate geometry.

The purpose of the present investigation is to provide a similar correlation or correlations for a three-dimentional turbulent boundary layer on the plate. The tripping wires at the edges of the flat plate are used to ensure that a turbulent boundary layer prevailed over the plate. The wooden sharp edge at the plate front is also used to decrease the bluntness effect of the plate. The experiments for determining average Nusselt numbers have been conducted for constant surface temperatures with Reynolds numbers ranging from 6.4×10^4 to 2.4×10^5, angles of attack varying from 0 to 45 deg, and angles of yaw from 0 to 30 deg

TEST RIG AND PROCEDURE

An experimental test rig is designed and constrected for the planned experiments, figure (1). It consisted of a low-turbulence open circuit wind tunnel of a 30-cm-square cross section, in which air from the laboratory room is drawn through the system, by a downstream blower (13). The flow rate is controlled by a throttle value (15). The velocity of the air stream drawn through the system is sensed by the help of Pitot tube (6) and an inclined alcohol manometer (26) at the centers of nine imaginary equal square areas into which its plane is divided. The Pitot tube being situated 30 cm upstream. The air velocity is also measured by the hot-wire probe (5) located 75 cm upstream. The difference in velocity values by the two methods is about $\pm$ 1% . Figure (1) shows the general layout of the experimental apparatus with the associated air supply system and heating plate. A schmatic diagram of the test section plate is shown in Fig. (2). A polished aluminum plate (11) having dimensions 150 mm long x 120 mm widex 2 mm thickness is used. 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 temperature is obtained as followes

$$Tm = \sum_{1}^{n} Tn / n, \qquad (6)$$

The temperatures measured 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 surface temperaturs and the measured values found to be in general less than 0.025 °C. The plate is heated electrically by means of the main heater (10). This heater consisted of a nickel chromiums heating wire wound around between two sheets of mica (9) each of 150 mm long x 120 mm wide x 1.5 mm thickness . The use of an aluminium test plate coupled with a closely wound heater ensured that the desired condition of a contant surface temperature is obtained. To prevent the heat loss from the main heater back, a guard heater (5) is used. The combination of the gaurd heater is mostly the same as the one used for the main heater is shown in Fig. (2). The heat input to each of the main and guard heaters is controlled by using two outo-transformers (18) and (21) as wall as two voltmeters (19 and 22) and two ammeters (20) and (23), see Fig. (1). A Bakelite plate (7) of dimensions 150 mm long x 120 mm wide x 16 mm thickness sandwitched between two aluminium sheets, each of dimensions 150 mm long x 120 mm wide x 2 mm thickness, and the whole set is placed between the set of the main and the gaurd heaters as shown in Fig. (2). For a fixed main heater input, the guard heater input is regulated so as to muintain as small a temperature difference as possible less than 0.2 °C ,

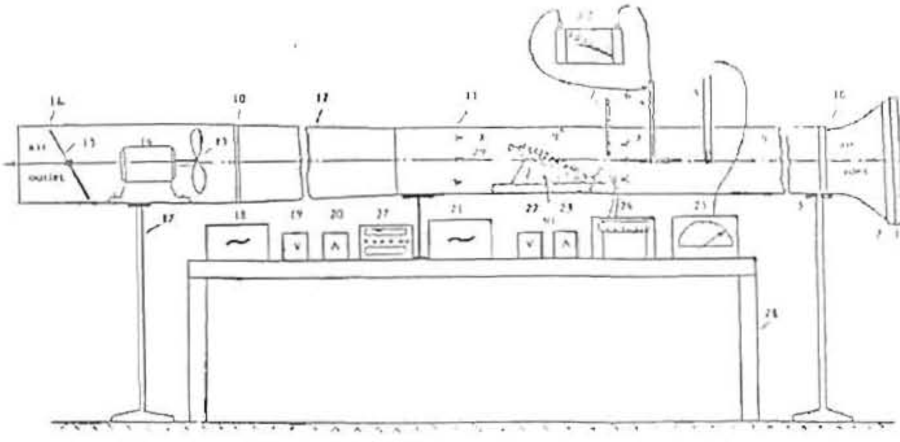

Fig. (1) Experimental test rig.
1. Stabilizing plate, 2. inlet collector, 3 and 17. windtunnel carriers, 4. stabilizing section, 5. thermoanemometer probe, 6. Pitot tube, 7. thermometer, 8. copper-constantan thermocouples, 9. test plate, 10. honeycomb, 11. test section, 12. connection section, 13. down stream blower, 14. electric motor, 15. throttle valve, 16. outlet section, 18. autotransformer of main heater, 19. voltmeter of main heater, 20. ammeter of main heater, 22. voltmeter of guard heater, 23. ammeter of guard heater, 24. temperature recorder, 25. thermoanemometer, 26. inclined manometer, 27. digital multimeter, 28. stand table 29. wooden wedge, 30. through bolt.

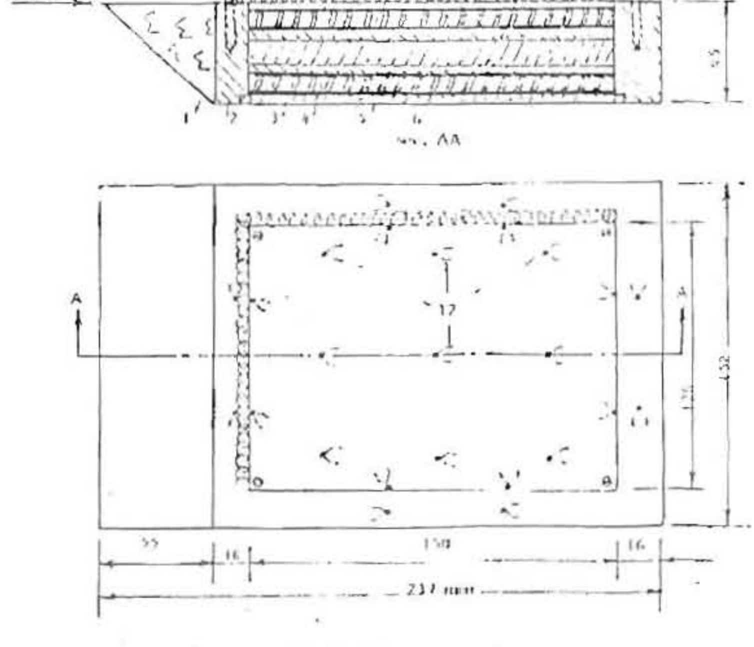

Fig. 2. Plate test section

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 , 13. tripping wire.

i.e. less than 0.01 mv on the digital multimetre (27) in Fig. (1), across the Bakelite plate thereby ensuring that the heat flow from the plate bottom is negligible. Eight thermocouple junctions are used for this purpose with four junctions on each side of the Bakelite plate. The junctions are located at the centers of four imaginary equal rectangular areas into which the plate sides are divided. These thermocouples are connected to digital multimeter (27) with an accuracy of 0.001 mv. Bakelite strips (16 mm thick) are fixed on all the four edges of the test plate (Fig. (2)). A wooden sharp edge (1) is fixed at the plate leading edge. In order to ensure a turbulent boundary layer, tripping wires (13) of 0.9 mm diameter are fixed on all edges having an incoming flow. 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 deep in the Bakelite frame and for 10 mm from its outer surface. On the outer surface of the Bakelite frame eight thermocouples are placed and embedded at the frame mid-height (Fig. (2)). The purpose of these thermocouples is to determine the heat loss by conduction (q_s) from the heating plate sides. The flow air stream temperatures before and after the test plate, and the wind tunnel surface temperature are measured by the set of thermocouples (8) located as shown in Fig.(1). The test plate assembly is supported on an adjustable wooden angle stand within the wind tunnel and fixed to its bottom by a through bolt (30) at its center. The wooden stand adjusted to fix the angle of attack (α) at 0 deg and by the help of wooden wedges it is fixed at 15, 30 and 45 deg and that of yaw (ψ) is adjusted by rotating the whole set around the through bolt at 0, 15, 30 and 45 deg, Fig. (1).

In order to find the heat lost by radiation (q_r) the average value of emissivity of 0.20 for polished aluminium plate is taken from [3], in which it was reported that no significant dependence of emissivity on temperatures was observed. The summation of estimated values of q_s and thoses of q_r are of order 15 to 35% 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 two and half hours. The average convective heat transfer coefficient is determined from the expression

$$h = (q_{in} - q_s - q_r - q_b) / A (T_s - T_0) \qquad (7)$$

The probable error in finding the average heat transfer coefficient is estimated to be about $\pm$ 9%.

It is found that the maximum blockage of the wind tunnel free stream cross section area at higher values of α and ψ is about 32.5%. Test and Tessmann [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.

RESULTS, DISCUSSION AND CONCLUSION

For the determination of the wind heat transfer coefficients and their correlation with airflow, some quantities are measured for each date run. These quantities are; the power input to the main heater, the heat lost by conduction from the test plate sides, the heat lost by radiation, the average plate surface temperature, the airflow stream velocity and the free stream temperature. The heat lost due to lacking balance between the main and guard heaters is neglected because the temperature difference on the sides of the Bakelite plate placed between the two heaters is kept very small (less than 0.2 °C). Net rate of heat transfered by convection is used to calculate the average heat transfer coefficient from equation (7). Because the flow is a three-dimensional flow, the characteristic dimension used in the calculation of both Nusselt (Nu_{L^*}) and Reynolds (Re_{L^*}) numbers is L* defined by L* = 4A/C.

During the course of experiments, the parameters varied independently included the Reynolds number, the angle of attack, the angle of yaw and the temperature difference. The Reynolds number ranging from 64000 to 240000, the angles of attack varying from 0 deg to 45 deg and the angle of yaw from 0 deg to 45 deg. The main stream velocity ranged from 9.2 to 27 m/sec. The temperature difference between the test plate surface and the oncoming air (ΔT) is varied from 9 to 33 °C at the given Reynolds numbers. In all, 143 data points are obtained and in addition, a number of experiments are repeated.

As mentioned before, the test section has a wooden sharp edge at the plate front to decrease the bluntness effect over the heated face and it has also tripping wires fixed on all edges having an incoming flow, in order to ensure a turbulent boundary layer. In such a case, the existance of the turbulent boundary layer is more realistic, because most of the engineering applications usually face some buildings, trees and some other constructions in which it may cause the turbulent flow type.

The results in terms of Nusselt number (Nu_{L^*}) versus Reynolds number (Re_{L^*}) for angles of attack equals O , 15 , 30 and 45 deg are shown in Figs. (3-6) respectively for all angles of yaw. No significant effect was noted, indicating that natural convection effects were negligible. This fact was also confirmed by using the criterion

$$\frac{Gr}{Re^2} < 0.002 \qquad (8)$$

The variation of heat transfer coefficients as seen in Figs. (3 - 6) will now be discuss ed . From the figures, it may be noted that the effect of yaw is not seen to be significant. This is probably due to the fact that with changes in yaw up to 30 deg, approximately the same average length of flow path is maintained. However, some effect have been observed with ψ equal to 45 deg, in which the figures show a little increase in the Nusselt number values. This observation is ingeneral existing except in the case of α equal to 45 deg, in which the Nusselt number values appear to be the lowest values in Fig. (6). This may be due to the fact that as the angle of attack is increased, the escape of flow past the lateral edges reduces the average length of flow path. In the case of angle of attack equal to 45 deg and angle of yaw equal to 45 deg, the pressure gradient increases and probably may set the process of relaminarization.

Inspection of Figs (3 - 6) show that the highest Nusselt number values are ingeneral reported with the case of angle of attack equal to O deg, in which the average length of the flow path is maximum and the escape of air from the lateral edges is minimum as seen in Fig.(3)

From the comparison between the reported data in Figs.(3 and 4), one may observe that the data displayed in Fig.(4) are much lower than the same reported in Fig.(3) by about 30%, i.e. as the angle of attack is increased to 15 deg Fig. (4), Nusselt numbers are lower than those obtained for α = O deg. This may be .

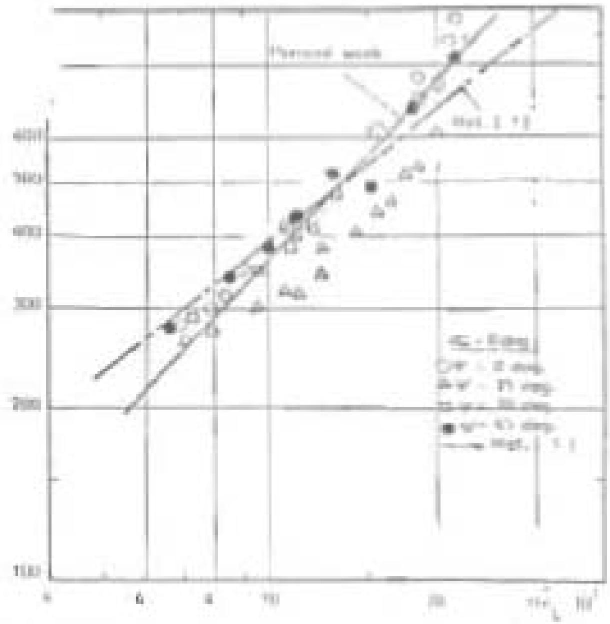

Fig. (19) Variation of Nusselt number with Reynolds number, angle of attack equal 0 deg.

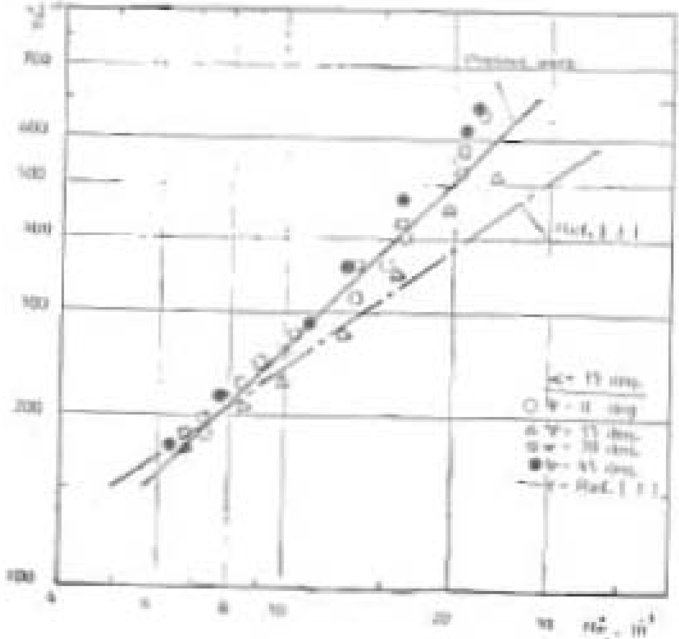

Fig. (20) Variation of Nusselt number with Reynolds number, angle of attack equal 15 deg.

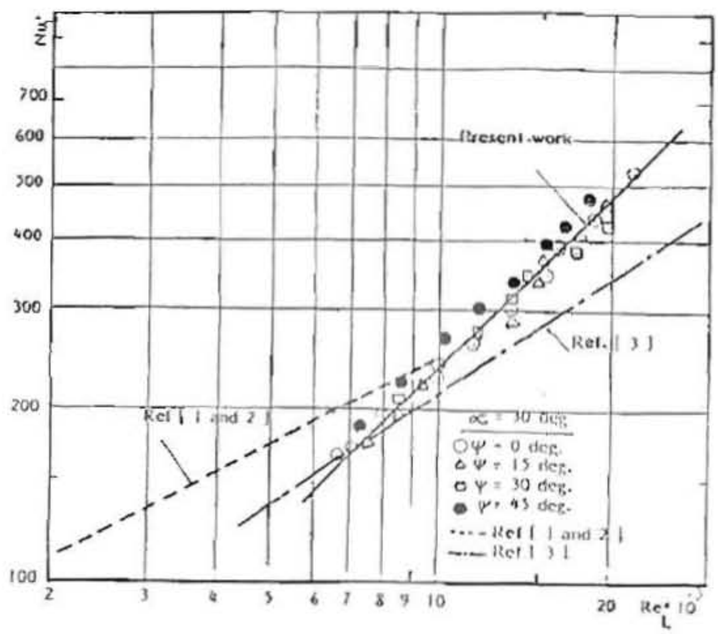

Fig . (5) Variation of Nusselt number with Reynolds number , angle of attack equal 30 deg.

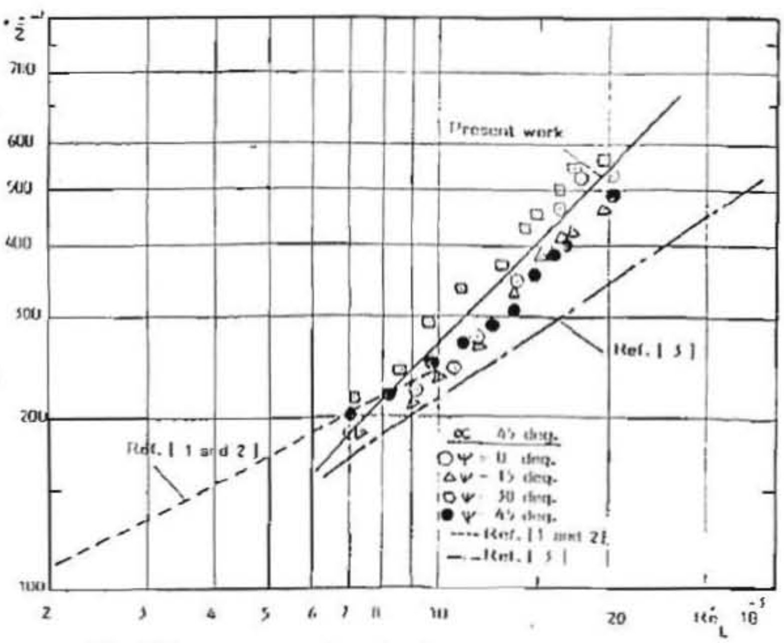

Fig . (6) Variation of Nusselt number with Reynolds number , angle of attack equal 45 deg.

as the angle of attack is increased, the average length of the flow path decreased due to the escape of flow past the lateral edges. Superimposed on this is the foundation of the separation bubble on the front portion of heated plate which reduces the rate of heat transfer from the top plate surface.

With the angle of attack further increased to 30 deg (Fig.5), it is expected that the separation bubble is going to vanish and a favorable pressure gradient probably sets in the process of relaminarization. This causes the Nusselt numbers to be lower than those for attack angle equal to 15 deg. In Fig.(6) one may observe that as the angle of attack increases to 45 deg the relaminarization effects increase also, but at the same time the average length of flow path reduces due to the increase of lateral escape, thus neutralizing the relaminarization effect.

COMPARISON WITH EXPERIMENTAL DATA

Overall inspection of Figs. (3 - 6) shows that the present experimental results, for each case, have been displayed in the Figures. The solid lines in the figures are obtained from the least square fitting of the experimental data. The figures also contain the solid lines obtained for the corresponding cases in Motwani's work [3]. Inspite of the present results and Motwani's results are obtained in case the tripping wires were fixed on all edges having an incoming flow, the present data appears to be slightly higher than the same of Motwani's work. This may be due to the foundatation of the wooden sharp edge at the plate front, in which it reduces the bluntness effect over the heated plate surface .

One may also observe in Figs (5 and 6) that equation (1) of Sparrow intersects with the present data at Re equal to 10^5. The similarities in trends have been obtained in spite of the differences in flow patterns between the two situations (the flow pattern is laminar in the case of Sparrow's work).

CORRELATIONS

Finally an attempt is made to correlate the results obtained in the present work. Such correlations are quite useful from a designer standpoint. The average Nusselt number (Nu_L^*) is correlated with the other relevant governing parameters, namely Reynolds number (Re_L^*) , angle of attack (α) and angle of yaw (ψ). The following correlation is obtained.

$$Nu_L^* = 0.535 \cdot \frac{(1+\psi)^{0.036}}{(1+\alpha)^{0.62}} \cdot Re_L^{*\,0.568} \qquad (9)$$

where

$$6.4\times10^4 \leq Re_L^* \leq 2.2 \times 10^5 ,$$

$$0 \text{ deg} < \alpha < 45 \text{ deg} , \text{ and}$$

$$0 \text{ deg} < \psi < 45 \text{ deg} ,$$

where α and ψ are in radiant. The correlation (9) predicts values of Nu_L^* which agree with results to within 3.5%. One may observe that the exponent of the term $(1+\Psi)$ is comparatively very small (0.036). This means that the results obtained are not sensitive to the angle of yaw. In fact this conclusion agree with that made by Motwani [3] and Sparrow [1 and 2] . As such the present results are again correlated , in case the angle of yaw is not included in the correlation as follows

$$Nu_L{}^* = 0.537 \ Re_L{}^{*0.569} \ (1+\alpha)^{-0.62} \qquad (10)$$

where

$$6.4 \times 10^4 < Re_L{}^* < 2.2 \times 10^5,$$

$$0 \text{ deg} < \alpha < 45 \text{ deg} ,$$

where α is in radiant. The correlation (10) predicts values of $Nu_L{}^*$ which agree with results obtained to within 5.6 % .

NOMENCLATURE

A	Plate area , m^2
Ap	projected area of the test plate , m^2
a	constant
b	constant
C	circumference of the plate , m
Gr	Groshof number , $g\beta\Delta t\, l^3 * /\nu^2$
h	average heat transfer coefficient , W/m^2K
K	thermal conductivity , W/m K
L	Plate length , m
L^*	characteristic length , 4A/C , m
Nu	Nusselt number , h L/K
$Nu_L{}^*$	Nusselt number , h L^*/K
Pr	Prandtl number , ν/α
q_b	heat loss from the plate bottom , W
qin	power input to test plate , W
qr	heat loss by radiation from the test plate , W
qs	heat loss from the plate sides , W
Re	Reynolds number , $u_\infty L/\nu$
$Re_L{}^*$	Reynolds number , $u_\infty L^* /\nu$
St	Stanton number , $h/\rho C p u_\infty$
Ts	average plate surface temperature , °C
T_∞	free - stream temperature , °C
ΔT	temperature difference , $(Ts - T_\infty)$, °C
u_∞	local average free stream velocity , m/s
α	angle of attack , thermal diffusivity , deg , , m^2/s
β	volumetric coefficient of thermal expansion $\frac{1}{K}$
ν	kinematic viscosity , m^2/s
ρ	density , Kg/m^3
Ψ	angle of yaw , deg .

REFERENCES

1- Sparrow , E.M., and Tien, K. K. , "Forced Convection Heat Transfer at an Inclined and Yawed Square Plate - Application to Solar Collectors" , Journal of Heat Transfer , Trans ASME, Vol. 99, p 507 (1977).

2- Sparrow, E. M., Ramsey, J. W. and Mass, E. A., "Effect of Finite Width on Heat Transfer and Fluid Flow About an Inclined Rectangular Plate", Journal of Heat Transfer, Trans. ASME, Vol. 101, p. 199 (1979).

3- Motwani, D. G., Gaitonde, U. N. and Sukhatme, S. P., "Heat Transfer from Rectangular Plates Inclined at Different Angles of Attack and Yaw to an Air Stream", Journal of heat transfer, Trans. ASME, Vol. 107, p. 307 (1985).

4- Shalaby, M. A., Araid, F. F. and Desoky, A. A., " Forced Convection Heat Transfer at an Inclined and Yawed Rectangular Plate" , Bulletin of the Faculty of Engineering, EL-Mansoura University, Vol.11, No. 1, PP. M. 27-M. 39, June, (1986).

5- Jurges, W., "Der Wäweübergang an einer ebenen wand", Beiheft. Zun Gesundheits-Ingenieur, Reihe I, Bicheft 19, (1924).

6- Test, F.L., and lessmann, R.C., "An Experimental Study of Heat Transfer During Forced Convection Over a Rectangular Body, "ASME Journal of Heat Transfer. Vol. 102, pp. 146-151, (1980).