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Experimental investigation of natural convection inside horizontal elliptic tube with different angles of attack

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Abstract

Natural convection from the inside surface of a horizontal elliptic tube of axis ratio (major to minor axis) 2:1 with a uniformly heated surface was investigated experimentally. The effect of angle of attack on the heat transfer rate was studied. The angle of attack (α) was varied from 0 (when the major axis was horizontal) to 90 (when the major axis was vertical) with steps of 15. The experiments covered a range of Rayleigh number Ra from 1.45×10^6 to 1.78×10^7. The local and average heat transfer coefficients and Nusselt number were estimated for different angles of attack at different Rayleigh numbers. The results obtained showed that the effect of angle of attack on the heat transfer coefficient is significant. The temperature distributions increase with the increase of axial distance from both ends of the tube until a maximum value at the middle of the tube. Also, the local Nu increases with the increase of α at the same axial distance. The average Nu_m increases with the increase of α at the same Ra. The results obtained were correlated by dimensionless groups and with the available data of the horizontal elliptic tube.

Keywords: Natural convection; Constant heat flux; Horizontal elliptic tube; Angle of attack

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Nomenclature

a, b	tube major and minor axis
C	constant
c	specific heat of air at constant pressure
D	hydraulic diameter of ellipic tube
D_i	inner diameter of tube
D_o	outer diameter of tube
g	gravity acceleration
Gr	Grashof number, $g\beta q\ D^4/k\nu^2$
h	local heat transfer coefficient
h_m	average heat transfer coefficient
k	thermal conductivity
L	elliptic cylinder length
Nu	local Nusselt number
Nu_m	average Nusselt number
Pr	Prandtl number, c μ/k
Q	total heat flow
q	heat flux, $Q/(\pi D\ L)$
Ra	Rayleigh number
s	circumferential distance measured from elliptic tube edge
t_a	ambient air temperature
t_w	inside wall temperature of elliptic tube
t_{wm}	average inside wall temperature of elliptic tube
t_{mf}	mean film temperature
X	axial distance measured from elliptic tube entrance.

Greek letters

α	angle of attack, equals zero when major axis of elliptic tube is horizontal
β	volumetric coefficient of thermal expansion
μ	dynamic viscosity of air
ν	kinematic viscosity of air, μ/ρ
ρ	density of air

1. Introduction

Natural convection heat transfer has gained considerable attention because of its employment in many practical fields in the area of energy conservation, design of solar collectors, heat exchangers, nuclear engineering, cooling of electrical and electronic equipments and many others.

The increasing interest in developing compact and highly efficient heat exchangers motivated researchers to study heat transfer from tubes of non-circular cross section. Tubes of elliptic cross section have drawn special attention since they were found to create less resistance to the cooling fluid, which results in less pumping power. Also, the elliptic tubes geometry is flexible enough to approach circular tubes when the axis ratio approaches unity and to approach a flat plat when the axis ratio is very small.

However, most of the available investigations are concerned with circular tubes. To the author's knowledge, there is a limited amount of experimental data on elliptic tubes concerned with natural convection from the outside surface. Raithby and Hollands [1] studied the problem of natural convection from an elliptic cylinder with a vertical plate and a horizontal circular cylinder as special cases. The study was limited to the vertical major axis configuration. Merkin [2] studied the symmetrical case of the same problem when either the major axis or the minor axis was vertical. The study was based on the solution of the boundary layer equations, and results were obtained for the entire surface, excluding the buoyant plume region. Hung and Mayinger [3] made an experimental study of natural convection from elliptic tubes with different orientations and with different axis ratios. A correlation of the average Nusselt number was reported. Badr and Shamsher [4] solved the problem of free convection from an elliptic cylinder for Rayleigh numbers ranging from 10 to 10^3 and axis ratios ranging from 0.1 to 0.964. The solution covered the entire flow region with no boundary layer approximations. Badr [5] studied laminar natural convection from an isothermal elliptic tube with different orientations. The tube orientation varies from horizontal to vertical major axis, while the axis ratio varies from 0.4 to 0.98. The study revealed that the average Nusselt number is a maximum when the major axis is vertical. Numerical theoretical and experimental investigations have been made on natural convection heat transfer in annuli. However, most of these investigations are concerned with enclosed spaces. The problem of natural convection inside horizontal and vertical annuli was studied by many investigators [6–9]. Khamis [7] and Al-Arabi et al. [8] investigated both theoretical and experimental studies of natural convection heat transfer inside vertical annuli; the theoretical study was for radius ratios of 0.26, 0.5 and 0.9 with two cases of heating the inner or outer tube while the other tube was adiabatic. Shehata [9], studied natural convection inside vertical, inclined and horizontal annuli of radius ratio of 0.73 with the inner tube heated and the outer tube adiabatic. Al-Arabi et al. [10] studied heat transfer by natural convection from the inside surface of a uniformly heated tube at different angles of inclination. The experiments were conducted in the range of Ra from 1.44×10^7 to 8.85×10^8, L/D from 10 to 31.4 and angle of inclination (measured from vertical position) from 0 to 75. The results showed that the average Nu_m was a maximum when the tube was vertical. The problem of natural convection inside open ended horizontal and vertical annuli with different annulus ratio was studied experimentally by Sarhan et al. [11]. Correlations of the dimensionless group of average $Nu_m - Ra \times D/L$ were presented for the horizontal and vertical cases.

As can be seen from the previous work, to the author's knowledge, there is no available data on natural convection inside horizontal elliptic tubes. The present work has been conducted to provide experimental data on natural convection heat transfer in open ended horizontal elliptic tubes with a uniformly heated wall and with different angles of attack.

2. Experimental apparatus and procedure

The experimental apparatus used is shown diagrammatically in Fig. 1. It consists essentially of a horizontal elliptic tube (test section) mounted on a frame that has a capability to rotate the test

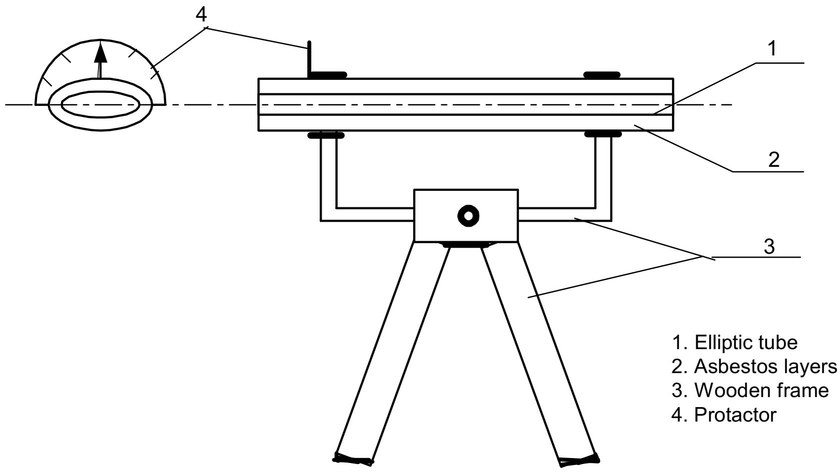

Fig. 1. Experimental setup.

section around its horizontal axis. The rotation angle α (angle of attack) can be read from a protractor, which was mounted on the test section. The elliptic tube was a copper one of 82 mm major axis, 41 mm minor axis and a thickness of 2 mm with 500 mm length. The outer surface of the elliptic tube was covered with an electric insulating tape on which nickel–chrome wire of 0.4 mm was uniformly wound to form the main heater as shown in Fig. 2. The main heater is covered with an asbestos layer of 55 mm thickness on which another nickel–chrome wire of 0.4 mm was uniformly wound to form a guard heater. The guard heater was covered with a 40 mm thick asbestos layer. Two pairs of thermocouples were installed in the asbestos layer between the main heater and the guard heater. The thermocouples of each pair were fixed on the same radial line. The input to the guard heater was adjusted so that, at steady state, the readings of the thermocouples of each pair became practically the same. Then, all the heat generated by the main heater is flowing inward to the elliptic tube.

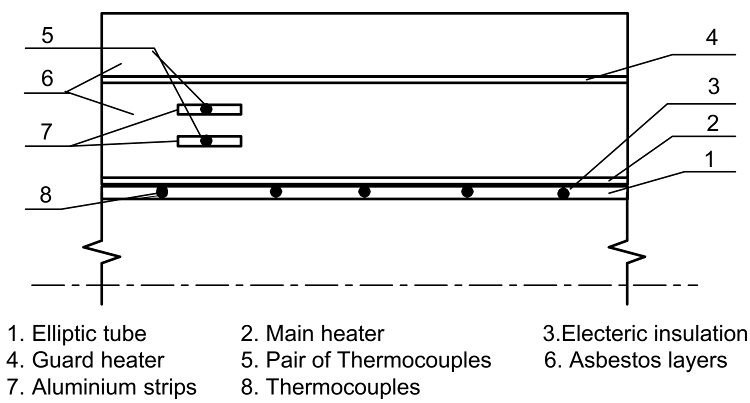

Fig. 2. Heaters arrangement.

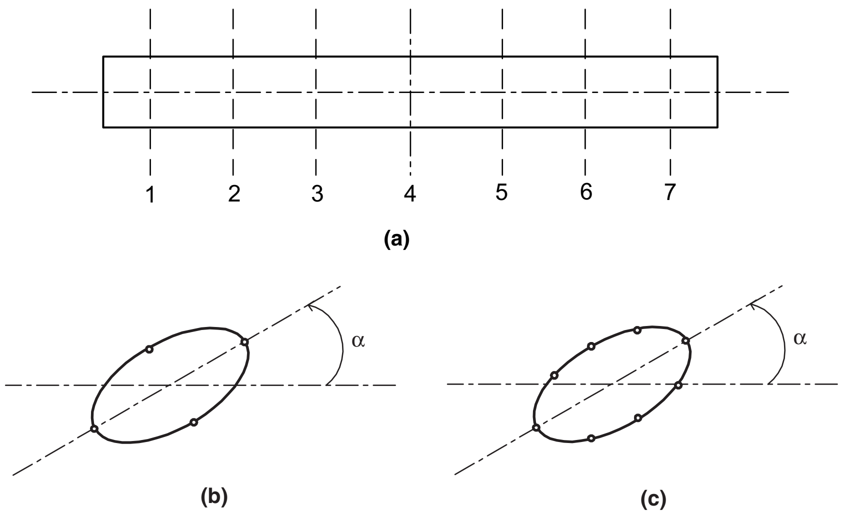

Fig. 3. Distribution of the thermocouples on the elliptic tube. (a) Thermocouples axial locations, (b) axial locations no. 1, 2, 3, 5, 6, 7 and (c) axial locations no. 4.

The inner surface temperature of the elliptic tube was measured by 32 copper–constantan thermocouples of 0.4 mm diameter that are soldered in slots milled in the axial and circumferential directions. The distribution of thermocouples on the inner surface of the elliptic tube was at seven measuring stations at axial distances of 50, 120, 190, 250, 310, 390 and 450 mm from one end of the elliptic tube as shown in Fig. 3a. Eight of the thermocouples are distributed on equal circumferential distances at 250 mm (middle of elliptic tube) and four of the thermocouples are distributed on each of the other six stations at equal circumferential distances as shown in Fig. 3b.

The readings of the thermocouples were taken by means a precalibrated digital temperature reader with a resolution of 0.01 °C. The ambient air temperature was measured by a precalibrated mercury-in-glass thermometer graduated in 0.1 °C. The input electric power was regulated by AC power variance and was measured by a digital wattmeter with a resolution of 0.01 W.

The experiments were conducted with the test section in a closed room 2.2 m × 2.2 m to prevent currents of air due to movements of anyone, and the measuring instruments were mounted outside of this room. The input electric power to the main heater was controlled and changed by the AC variac at each angle of attack. The angle of attack was changed from zero to 90° with steps of 15°. The steady state condition for each run was achieved after 3–4 h approximately. The steady state condition was considered to be achieved when the difference of the wall temperatures were not changed by more than 0.5 °C within 20 min. When the steady state condition was established, the readings of all thermocouples, the input power and the ambient temperature were recorded.

3. Uncertainty analysis

Generally, the accuracy of the experimental results depends upon the accuracy of the individual measuring instruments and the manufactured accuracy of the elliptic tube. Also, the accuracy of an instrument is limited by its minimum division (its sensitivity). In the present work, the uncertainties in both the heat transfer coefficient (Nusselt number) and Rayleigh number were estimated following the differential approximation method.

For a typical experiment, the total uncertainty in measuring the main heater input power, temperature difference ($t_w - t_a$), heat transfer rate and the elliptic tube surface area were ±0.4%, ±0.75%, ±1.2% and ±0.5%, respectively. These were combined to give a maximum error of ±3.2% in heat transfer coefficient (Nusselt number) and a maximum error of ±4.5% in Rayleigh number.

4. Results and discussion

4.1. Definitions

In the present work, the local heat transfer coefficient between the inner surface of the elliptic tube and the air inside the elliptic tube is calculated from Eq. (1):

$$h = q/(t_w - t_a) \tag{1}$$

The corresponding local Nusselt number, Nu is calculated from Eq. (2):

$$Nu = h \times D/k \tag{2}$$

The average heat transfer coefficient between the inner surface of the elliptic tube and the air inside the elliptic tube is calculated from Eq. (3):

$$h_m = q/(t_{wm} - t_a) \tag{3}$$

where

$$t_{wm} = \int_0^l t_w \, dx \tag{4}$$

The corresponding average Nusselt number Nu_m is calculated from Eq. (5):

$$Nu_m = h_m \times D/k \tag{5}$$

The Rayleigh number Ra is calculated from Eq. (6):

$$Ra = Gr \times Pr \tag{6}$$

The physical properties are evaluated at the mean film temperature, Ref. [12], as given by:

$$t_{mf} = (t_{wm} + t_a)/2 \tag{7}$$

4.2. Temperature distribution on the inside surface of the elliptic tube

The measurements of the temperature on the inside surface of the elliptic tube for all the experiments showed that the only variation in temperature readings is axially, and there is no significant difference in the temperatures with the circumferential distance (s/a). Generally, the maximum difference in temperature readings with the circumferential distance (s/a) is ±1.5° at the same axial distance and heat flux.

Fig. 4 shows the variation of the temperature difference between the inside surface temperature of the elliptic tube and the ambient air temperature ($t_w - t_a$) along the axial length for the different heat fluxes employed and for the angle of attack of 45°. The ($t_w - t_a$)–X curves for all other radius ratios have the same general shape. The surface temperatures gradually increase from both ends of the elliptic tube until a maximum value is reached at the middle point of the elliptic tube. The general shape of all the curves gives symmetry around the middle point of the elliptic tube. The reason for this shape of the ($t_w - t_a$)–X curves is explained by the growing of the boundary layer in both ends along the axial distance.

Fig. 5 shows the effect of angle of attack (α) on the temperature difference ($t_w - t_a$) along the axial length of the elliptic tube. The temperature difference decreases with the increase of angle of attack (α) at the same axial distance. In general, the temperature difference shows maximum values at $\alpha = 0°$ and minimum values at $\alpha = 90°$, and the temperature difference is inversely proportional with the increase of α.

4.3. Nusselt number

Fig. 6 shows the variation of local Nusselt number (Nu) with axial distance of the elliptic tube for the different heat fluxes employed and for an angle of attack of 45°. The local Nusselt number (Nu) has a maximum value at both ends of the elliptic tube, and it decreases gradually until it reaches a minimum value at the middle of the elliptic tube. All the other cases of the Nu–X relations at the other angles of attack give the same general shape. This can be explained from the curves of both Figs. 4 and 5. The variation of Nu along the axial distance of the elliptic tube depends on the temperature distributions. The growing boundary layer from both ends toward the middle of the elliptic tube causes the increase of temperature distributions and consequently the decreases of the Nu towards the middle of the elliptic tube. Fig. 7 shows the effect of angle of

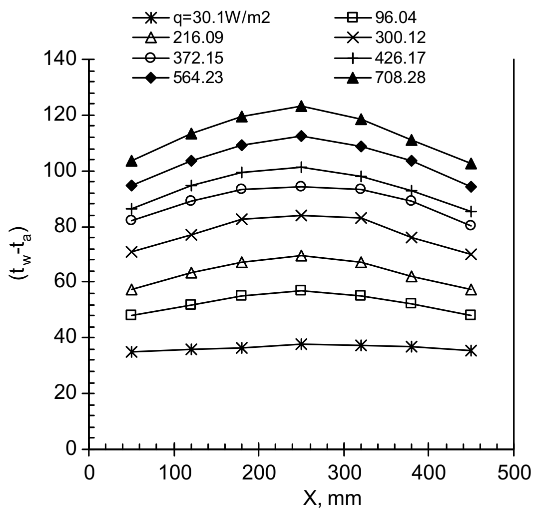

Fig. 4. Variation of ($t_w - t_a$) with X at $\alpha = 45$.

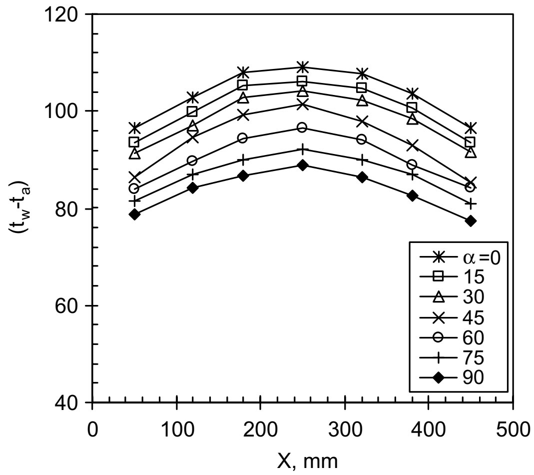

Fig. 5. Effect of α on ($t_w - t_a$) at $q = 426.17$ W/m^2.

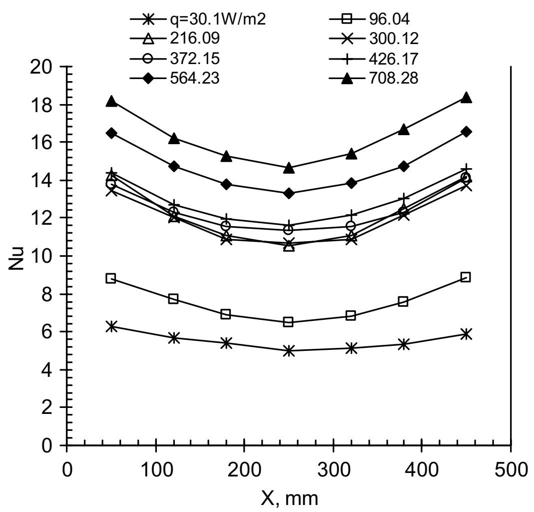

Fig. 6. Variation Nu with X at $\alpha = 45$.

attack (α) on the local Nu at $q = 426.17$ W/m^2. The local Nu increases with the increase of α at the same axial distance.

The variation of the average Nusselt number with Ra for all angles of attack is shown in Fig. 8. Nu_m increases with the increase of Ra for all angles of attack. A comparison between Nu_m for all angles of attack shows that Nu_m increases with the increase of α, Fig. 8.

4.4. Correlation of the results

The general relation between Nu_m and Ra for all angles of attack is given by:

$$Nu_m = CRa^n(1 + \sin\alpha)^m \tag{8}$$

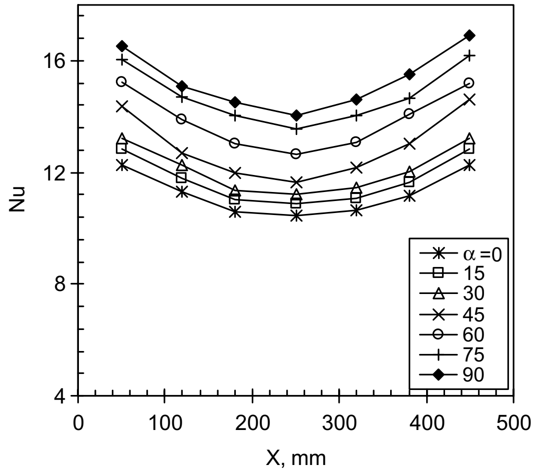

Fig. 7. Effect of α on Nu at $q = 426.17\ \mathrm{W/m^2}$.

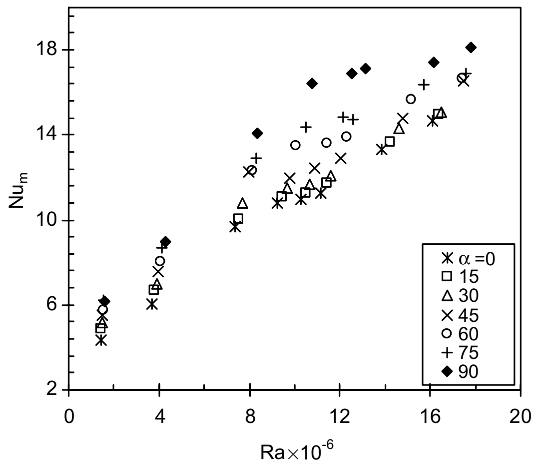

Fig. 8. Effect of α on the average Nu for all cases.

where n and m are constants. The experimental results were fitted using power regression to determine the constants. The resulting empirical correlations are:

$$Nu_{\mathrm{m}} = 0.067Ra^{0.32}(1+\sin\alpha)^{0.217}, \quad 1.45\times10^{6} \leqslant Ra \leqslant 1.78\times10^{7};\ 0^{\circ} \leqslant \alpha \leqslant 90^{\circ} \tag{9}$$

Eq. (9) represents the general equation for natural convection inside horizontal elliptic tubes.

The calculated data from Eq. (9) of the average Nusselt number (Nu_{mCal}) is plotted against experimental data of the average Nusselt number (Nu_{mExp}) in Fig. 9. As shown in this figure, the maximum deviation between the experimental data and the correlation of the inclined elliptic tube is ±14%.

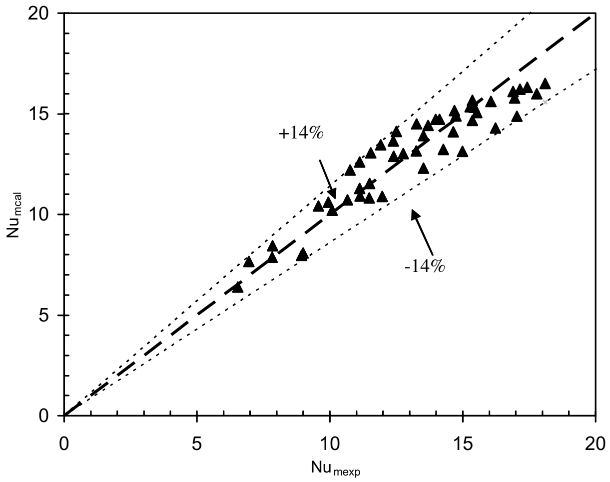

Fig. 9. Average Nu_{cal} vs Average Nu_{exp} for horizontal elliptic tube.

5. Comparison with the previous work

As mentioned before, to the author's knowledge, there is no published information about natural convection heat transfer inside horizontal elliptic tubes that can be compared with the present work. The only available information that can be compared with the present work was established by Sarhan et al. [11]. The comparison is made with the dimensionless groups of $Nu_{\text{m}} - Ra \times D/L$ of the present work with the work of Sarhan et al. [11] with annulus ratio of zero (plain circular tube) as shown in Fig. 10. As shown in this figure, the present results of natural convection inside horizontal elliptic tubes for all angles of attack are higher than the results of Sarhan et al. [11] for natural convection inside horizontal circular tubes.

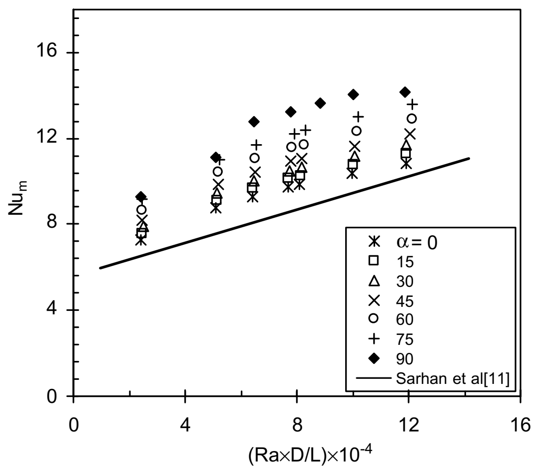

Fig. 10. Comparison of the present work with Sarhan et al. [11].

6. Conclusion

Natural convection heat transfer from the inside surface of the horizontal elliptic tube of axis ratio 2:1 with a uniformly heated surface was investigated experimentally. The angle of attack was changed from 0° to 90° with steps of 15°. The experiments covered a range of Rayleigh number, Ra from 1.45×10^6 to 1.78×10^7. The local and average heat transfer coefficients and Nusselt number were estimated for different angles of attack at different Rayleigh numbers. It was found that the angle of attack has a significant effect on the local and average heat transfer coefficients. The temperature distributions increase with the increase of axial distance from both ends of the tube until a maximum value at the middle of the tube. Also, the local Nu increases with the increase of α at the same axial distance. The average Nu_m increases with the increase of α. Correlations of Nusselt number for natural convection inside open ended horizontal tubes at different angles of attack were presented.

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