Free convection from a vertical and inclined semicircular cylinder at different orientations

S. A. Nada [a] and M. Mowad [b]
[a] Mech. Eng. Tech. Dept., Benha High Institute of Technology, Benha 13512, Egypt
[b] Mech. Eng. Dept., Faculty of Eng., Zagazig University, Shoubra, Egypt

Free convection heat transfer from the outside surface of a vertical and inclined semicircular cylinder was investigated experimentally at different orientations. The experiments were carried out to study the effect of the inclination angle and the orientation angle of the semicircular cylinder on the distribution of the wall temperature, local and average heat transfer coefficients and local and average Nusselt numbers for a wide range of Grashof numbers. Also, correlations for the coefficient of heat transfer by free convection from a vertical and inclined semicircular cylinder were obtained in a dimensionless form as a function of Rayleigh numbers, inclination angles, and orientation angles of the cylinder. The experiments were carried out at four inclination angles of the semicircular cylinder, namely at α = 0°, 30°, 45° and 60° measured from the vertical. For each inclination three different orientations of the semicircular cylinder were investigated: (1) the flat part of the semicircular cylinder is facing up (θ = 0°), (2) the flat part of the semicircular cylinder is vertical (θ = 90°), and (3) the flat part of the semicircular cylinder is facing down (θ = 180°). The experiments were conducted at constant heat flux. The results showed that: (i) the average Nusselt number increases as the inclination angle of the semicircular cylinder (measured from the vertical) increases for the three different orientations of the cylinder, and (ii) At any inclination of the cylinder, the Nusselt number for the orientation θ = 90° is higher than that of the orientation θ = 0° which in turn is higher than that of θ = 180°. Correlations were developed to predict the average Nusselt number of the semicircular cylinder in terms of the Rayleigh number, the inclination angle, and the orientation angle of the cylinder. The trend of the present results was compared with that available in the literature and fair agreement was found.

الغرض من هذا البحث هو تقديم دراسة معملية لعملية انتقال الحرارة بالحمل الطبيعي من نصف اسطوانة مائلة عند أوضاع مختلفة. وأجريت الدراسة لثلاث زوايا ميل مختلفة بالأضافة للوضع الرأسى. ولكل زاوية ميل تمت الدراسة عند ثلاث أوضاع مختلفة للأسطوانة: (أ) الجزء المسطح من نصف الأسطوانة أفقي ومواجه لأعلى, (ب) الجزء المسطح من نصف الأسطوانة أفقي ومواجه لأسفل, (ت) الجزء المسطح من نصف الأسطوانة رأسي. وأجريت الدراسة عند ثبات الفيض الحراري. وقد بينت النتائج: (أ) زيادة معدل انتقال الحرارة بزيادة زاوية الميل من الوضع الرأسى وذلك للأوضاع المختلفة للأسطوانة. (ب) لأى زاوية ميل كان الوضع التالث له أعلى معدل انتقال حرارة. وتم استنتاج معادلات معملية فى صورة لابعدية تعطى معامل انتقال الحرارة لزوايا ميل مختلفة وذلك للأوضاع الثلاثة. وقد قورنت النتائج بالأعمال السابقة.

Keywords: Free convection, Semicircular inclined cylinder

1. Introduction

Geometry and orientation of heat transfer surfaces are two major parameters affecting the rate of convection heat transfer from these surfaces. Therefore, results and correlations for a certain geometry and orientation of a heat transfer surface would not apply to other geometries or orientations. To our knowledge, no experimental data and correlations are available yet for free convection heat transfer from an inclined semicircular cylinder at different orientations. Such data and correlations are valuable in several applications in the area of heat transfer involving free convection. Examples of these applications include cooling of electronic devices and solar energy components. Knowledge of characteristics of heat transfer from this surface at different inclinations and orientations should guide the design of the devices used in these applications.

The process of heat transfer by free convection from the outside surfaces of a vertical and inclined semicircular cylinder at different orientations involves, in interconnecting manner, the processes of heat transfer by free convection from vertical flat plate, vertical cylinder, inclined flat plate facing down, inclined flat plate facing up and inclined cylinder.

For free convection from a vertical flat plate, expressions of the average Nusselt number were obtained by many previous authors (McAdams [1], Sparrow and Gregg [2] and Warner and Arpaci [3]) for different ranges of Rayleigh numbers. More Recently, Churchill and Chu [4] have recommended correlations for free convection from a vertical plate that may be applied for any value of Rayleigh number. Also, they showed that these correlations can also be used for vertical cylinder if the diameter of the cylinder is sufficiently large compared with its axial length. Also, Churchill [5] proposed a correlation for free convection heat transfer for vertical and inclined cylinder with end surfaces insulated for a certain range of Rayleigh number.

For free convection heat transfer from inclined flat plate, Suryanarayana [6] showed that: (i) for a downward-facing heated or an upward-facing cooled inclined isothermal surface the correlation for free convection heat transfer from a vertical flat plate can be used by using the component of the gravitational force parallel to the plate to evaluate Rayleigh number, (ii) for an upward-facing heated plate the fluid flow in free convection is more complex than with a downward-facing heated plate and new correlations are required. Fujii and Imura [7] studied experimentally the free convection from isothermal upward-facing heated inclined plate subjected to a uniform heat flux. The study showed that the boundary layer around the surface depends on Grashof number. Two correlations were presented for different ranges of Grashof numbers. For an inclined plate subjected to a uniform heat flux, with the heated surface facing down, Fussey and Warneford [8] recommended two correlations for Nusselt number with different ranges of inclination angle.

For horizontal flat plate, McAdams [1], Fujii and Imura [7], Goldstein et al. [9] and Lloyd and Moran [10] conducted experiments on free convection heat transfer from downward and upward-facing heated horizontal rectangular plates and correlations were presented for the two cases at different ranges of Rayleigh numbers. Churchill and Chu [11] and Morgan [12] presented comprehensive correlations for free convection from a horizontal cylinder for a wide range of Rayleigh number. Recently, Nada et al. [13] studied experimentally free convection heat transfer from semicircular horizontal cylinder at different orientations. Correlations of Nusselt number for different orientations of the flat part of the semicircular cylinder with respect to the vertical were presented.

Sparrow and Stretton [14] performed detailed natural convection experiments with cubes in many different orientations. From their experimental results and the data of previous researches (Churchill [5] and Raithby and Hollands [15]) with spheres and cylinders (radius equal to the axial length) they proposed a correlation to predict the average Nusselt number for free convection for bodies with unity aspect ratios (like cubes, spheres, cylinders of radius equal to the axial length). The validity of the proposed equation for other bodies is not yet established. Also, Lienhard [16] suggested a correlation for rough estimation of free convection Nusselt number for bodies for which correlations are not available.

It can be noted from the above discussion, that free convection from surfaces of various geometries and orientations (vertical plate, vertical cylinder, inclined plates and cylinders, horizontal flat plates, horizontal cylinders, semicircular horizontal cylinder and spheres) have been studied by previous investigators. That is because free convection heat transfer is strongly dependent on surface geometry and orientations, therefore, results on certain geometry and orientation would not apply to other geometries or orientations.

To our knowledge, experimental data are not available yet for free convection from inclined semicircular cylinder at different orientations. Therefore, the aim of the present investigation was to examine (experimentally) and correlate the free convection heat transfer characteristics from a semicircular inclined cylinder at different orientations. The study was carried out for different values of the independent parameters: Grashof number, inclination of the semicircular cylinder and the orientation angle of the cylinder.

2. Experimental setup and procedure

2.1. Experimental setup

The experimental setup is shown in fig. 1. It consists of a semicircular cylinder (test section) mounted on an inclination and rotatable frame to vary the inclination angle α and the orientation angle θ of the test section (see fig. 1-c). A measuring protractor is used to read these angles. The test section is made from copper and has 80-mm outside diameter and 3-mm thickness. The length of the test section is 480 mm. A wooden semicircular rod of diameter 60-mm and 480-mm length was inserted inside the test section. A flat electric resistance Nickel-Chrome wire was wrapped around the wooden rod to generate the required heat input. The electric heater wire is insulated from the bottom and the top with mica insulating tape. The gap between the wooden rod with the heater and the semicircular test section was filled with fine sand to be sure of uniform distribution of the input heat. The cross sectional view of the test section is also shown in fig. 1-b. The experimental set up was placed inside four sides plastic walls (the distance between each two walls was 3 m) to prevent any wind to affect on the results. The input power was regulated by an AC power variac and measured by a digital Wattmeter. Wall temperatures were measured using 28 thermocouples (T-type) mounted at seven axial locations. Each axial location contains 4 thermocouples equally distributed on the circumference of the test section. The first axial location is at 45-mm from one end of the test section. The next axial locations were 65-mm equally distance apart. The axial and circumferential distribution of thermocouples on the test section is shown in fig. 1-a and fig. 1-b, respectively. The thermocouples wires were extended through the gap between the semicircular cylinder and the wooden rod.

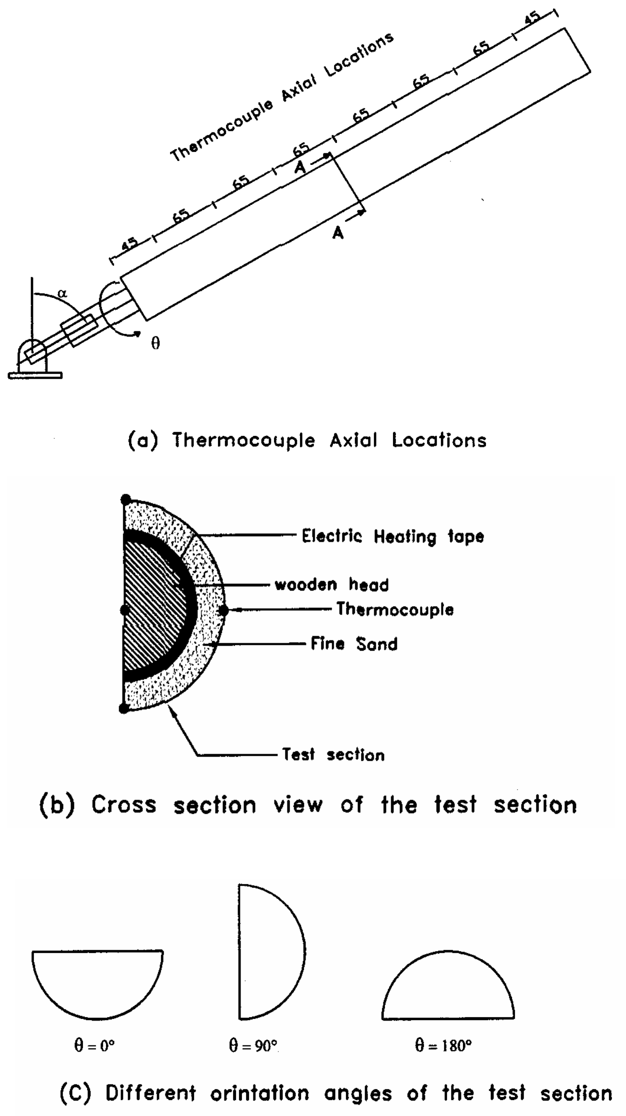


Fig. 1. Experimental setup.

2.2. Experimental procedure

The three independent variables in these experiments are the Grashof number (controlled by the input power), the inclination angle of the test section and the orientation angle of the test section. After adjusting the desired conditions of the experimental run, the experiment was allowed to run for at least 4 hours before steady state conditions were achieved. When steady state condition was established, the readings of all thermocouples, the input power and the ambient temperature were recorded by a data acquisition system.

The steady state condition was considered to be achieved when the change in the wall temperature was not more than 0.2 °C within 15 minutes. The rate of heat losses by radiation from the test section was calculated and subtracted from the input power to obtain the rate of heat transfer from the test section to the surrounding air by free convection. The percentage of the heat losses by radiation was found to be within 40% from the heat input for all test runs. Experiments were carried out for four inclinations of the test section α = 0°, 30°, 45°, and 60° and three orientations of the test section, θ = 0°, 90°, and 180°. A total of about 10 runs were carried out for each inclination and each orientation of the test section at different input powers.

3. Data reduction

The dimensionless interrelated independent variables *Gr* and *Ra* were calculated from the measured quantities using the following definitions:

$$Gr = \frac{\beta g \rho^2 L^4 q''}{\mu^2 k} \quad \text{and} \quad Ra = Gr\, Pr\,. \quad (1)$$

Where, q'' represents the average heat flux transferred by free convection from the surface of the test section to the surrounding air. To calculate q'', an energy balance for the heated test section gives;

$$VI = (\pi/2 + 1)\, D\, L\, (q'' + q_r''). \quad (2)$$

Where I and V are the electric current and the voltage input to the heating element and q_r'' is the radiation heat flux lost from the hot test section to the surrounding walls. The radiation heat loss q_r'' from the hot test section is estimated as;

$$q_r'' = \sigma\varepsilon\, [(T_s + 273)^4 - (T_\infty + 273)^4]\,. \quad (3)$$

Where T_s is the average temperature of the test section and ε is its emissivity. In the last equation a large enclosure is assumed.

After calculating q'', the local heat transfer coefficient was calculated from:

$$h_i = \frac{q''}{(T_i - T_\infty)}\,. \quad (4)$$

Where i refers to the wall thermocouple position on the axial location of the tube, as shown in fig. 1 and T_i is the circumferential averaged local temperature at this axial position. The average heat transfer coefficient along the surface of the tube is calculated in two ways:

(i) By determining the length average heat transfer coefficient along the length of the cylinder from:

$$\bar{h} = \frac{1}{L}\sum h_i \Delta x_i\,. \quad (5)$$

Where Δx_i is the axial distance element around the thermocouple of axial position number i.

(ii) By determining the average temperature along the surface of the tube (T_s) and using it to find the average heat transfer coefficient from:

$$\bar{h} = \frac{q''}{(T_s - T_\infty)}\,. \quad (6)$$

The two values obtained from (i) and (ii) were very close and, therefore, the average mean heat transfer coefficient was taken as the mean of these two values (Busedra and Soliman [17] and Nada et al. [13]) i.e.,

$$\bar{h} = \frac{1}{2}\left[\frac{1}{L}\sum_{1}^{7}\frac{q''\Delta x_i}{(T_i - T_\infty)} + \frac{q''}{(T_s - T_\infty)}\right]. \quad (7)$$

The value from the last equation will be called the average heat transfer coefficient in the following sections without using the word mean for briefness.

The local Nusselt number was calculated from the following definition:

$$Nu_i = \frac{h_i\, x_i}{k} = \frac{q''\, x_i}{k\,(T_i - T_\infty)}\,. \quad (8)$$

Where x_i is the axial distance measured from the lower end of the cylinder to the axial thermocouple position *i*. The average Nusselt number along the tube was calculated from:

$$\overline{Nu} = \frac{\overline{h}L}{k} . \quad (9)$$

All air properties in the last equations were calculated at the film temperature given by:

$$T_f = (T_s + T_\infty)/2 . \quad (10)$$

4. Experimental results and discussion

A total of 100 test runs were conducted in the present study covering the following ranges and values:

$Gr = 1.5 \times 10^9 - 8.5 \times 10^{10}$
$Pr = 0.695 - 0.707$ (air)
$Ra = 10^9 - 6 \times 10^{10}$
$\alpha = 0°, 30°, 45°$ and $60°$
$\theta = 0°, 90°$ and $180°$

4.1. Wall temperature

In all experiments, it was found that the circumferential variation of wall-temperature measurements at any axial location is small for any inclination and orientation of the test section. Generally, it was within ± 1 °C under constant heat flux conditions. This can be attributed to the high thermal conductivity of the tube and the low Prandtl number of the air, which may lead to relatively mild stratification in the circumferential direction.

The axial variation of the cylinder wall temperature for a certain experimental run is shown in fig. 2. As shown in the figure, the axial temperature of the cylinder increases with increasing the axial distance measured from the bottom of the cylinder. This can be attributed to the known shape of boundary layer development around vertical and inclined plates and cylinder, which begins at the lower end of the vertical surfaces and increases with increasing the axial distances from the lower end. The increase of the boundary layer thickness leads to the decrease of the heat transfer rate and thus an increase of the surface temperature for a constant heat flux condition. The figure shows that the temperature gradient in the axial direction deceases with increasing the axial distance from the lower end. This can be attributed to the increase of the turbulence of the boundary layer with the increase of axial distance. Increasing the turbulence leads to the increase of the heat transfer coefficient and therefore a decrease in the surface temperature. This trend of the axial temperature distribution is observed through out all the experimental runs.

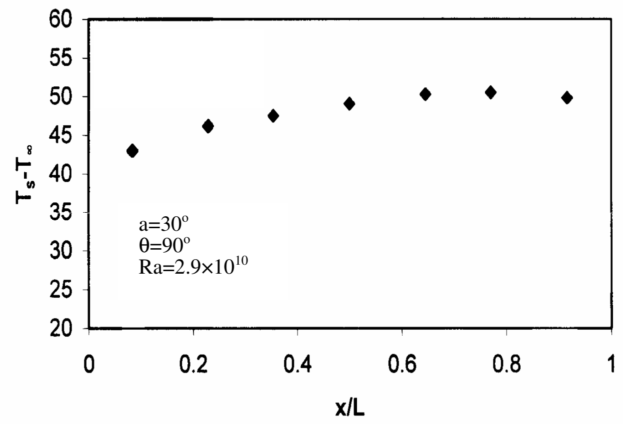


Fig. 2. Axial variation of test section wall temperature.

The variations of the average surface temperature of the semicircular cylinder with the Grashof number for different inclination and orientation angles are shown in fig. 3 and fig. 4, respectively. Fig. 3 shows that the average wall temperature decreases with the increase of the wall inclination angle. The trend is the same for all the orientation angles of the cylinder. Fig. 4 shows that the average surface temperature of the cylinder for the orientation θ = 90° is smaller than that of θ = 0° which in turn is smaller than that of θ = 180°. The trend is the same for all the inclination angles of the cylinder. This trend can be attributed to the decrease of the average boundary layer thickness around the cylinder with the increase of the inclination angle. Also the average boundary layer thickness for the orientation θ = 90° is smaller than that of θ = 0° which in turn is smaller than that of θ = 180°. The decrease of the boundary layer thickness increases the rate of heat transfer which leads to a decrease in the average wall temperature for constant heat flux condition.

4.2. Nusselt number

Fig. 5. shows the axial variation of the local Nusselt number for a certain experimental run. As shown on the figure, the local Nusselt number increases with the increase of the axial distance measured from the lower end of the cylinder. This can be attributed to the direct change of the local Nusselt number with the axial distance x. Therefore, the increase of x/L leads to an increase in local Nusselt number.

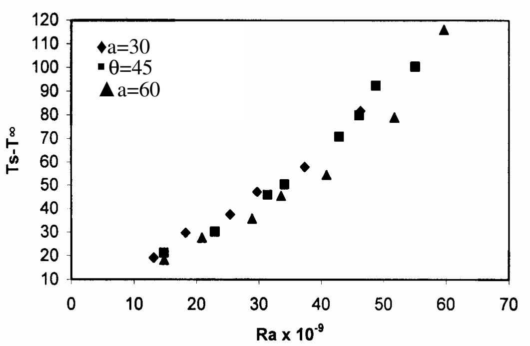


Fig. 3. Variation of test section average surface temperature with Ra for different inclination angles.

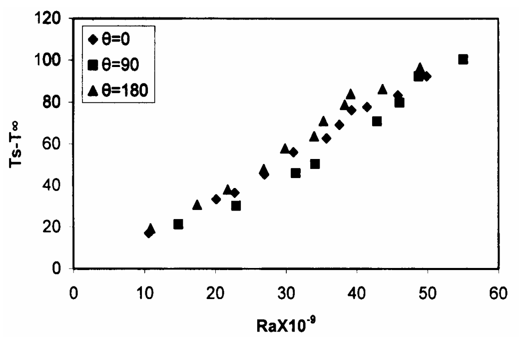


Fig. 4. Variation of test section average surface temperature with Ra for different orientation angles.

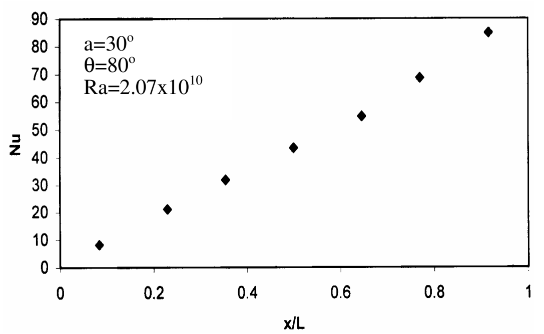
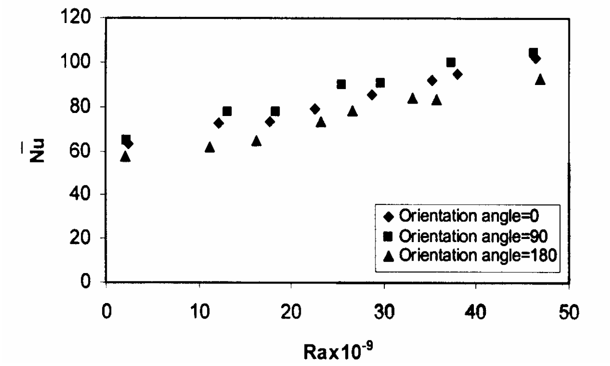
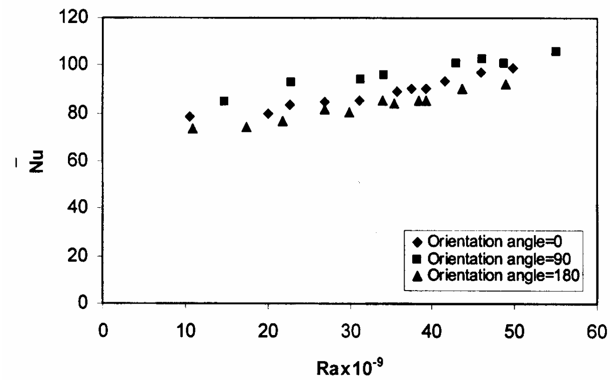


Fig. 5. Axial variation of local Nusselt number.

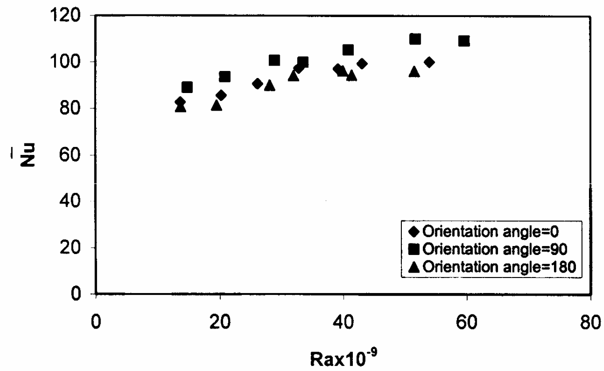
The effect of the orientation angle of the cylinder on the average Nusslet number is shown in fig. 6 for different inclinations of the cylinder. The figure shows that the average Nusslet number for the orientation θ = 90° is greater than that of θ = 0° which in turn is greater than that of θ = 180°. This can be attributed to the average boundary layer thickness around the surface of the semicircular cylinder. The average boundary layer thickness for the orientation θ = 90° is smaller than that of the orientations θ = 0° and θ = 180°. As the boundary layer thickness decreases the heat transfer coefficient and the Nusselt number increase. This variation of the average boundary layer thickness with the orientation angle can be explained in the following three points according to the view of the boundary layer shape around the surface of the semicircular cylinder for the different orientations: (i) The length of the boundary layer path along the surface of the semicircular cylinder for the orientation θ = 90°, which equals to $D/\sin\alpha$, is smaller than that of the orientations θ = 0° and θ = 180°, which equals to the length of the cylinder (the boundary layer start at x = 0 and is coincides along the length of the cylinder). This makes the average boundary layer thickness for the orientation θ = 90° smaller than that of the orientations θ = 0° and θ = 180°. (ii) For the orientation θ = 0°, the flat part is upward facing and the boundary layer detaches itself from the surface due to the component of the buoyant force perpendicular to the surface. The strength of this detachment increases with the increase of the turbulence of the boundary layer, i.e. with the increase of the Ra. While, for the orientation θ = 180°, the flat part is downward facing and the component of the buoyant force perpendicular to the surface is hindered by the surface and this makes the boundary layer moves parallel to the surface along the tube without any detachment. (iii) The known shape of the boundary layer development around complete horizontal cylinder, which begins at the bottom of the cylinder with small thickness and concludes at the top of the cylinder, leads to a smaller boundary layer thickness for the circular part of the semicircular cylinder in the case of θ = 0° than that of θ = 180°.



(a) Inclination angle =30°



(b) Inclination angle =45°

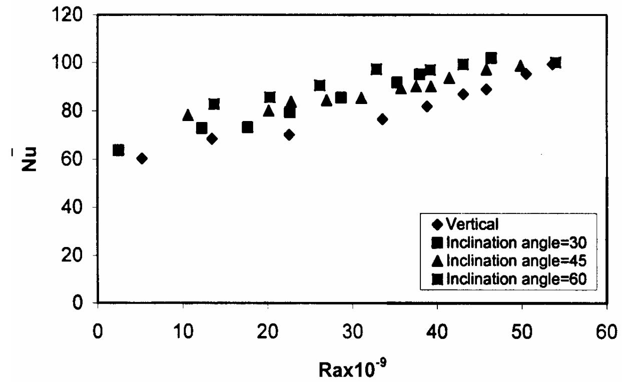


(c) Inclination angle =60°

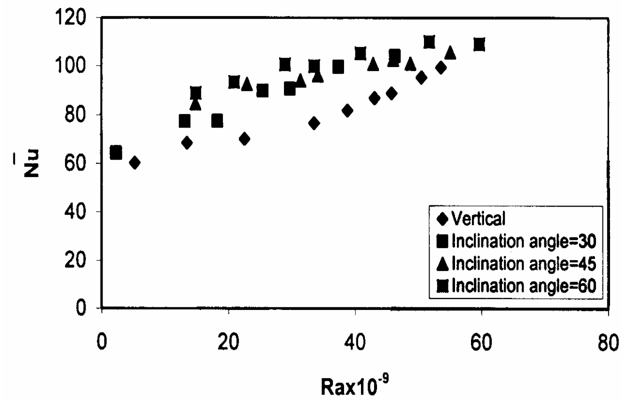
Fig. 6. Effect of orientation angle on average Nusselt number.

The effect of the inclination angle of the semicircular cylinder on the average Nusselt number is shown in fig. 7 for the different orientations of the cylinder. As shown in the figure, the average Nusselt number increases as the inclination angle of the cylinder increases. The trend is the same for the three orientations. This can be attributed to the following reasons:

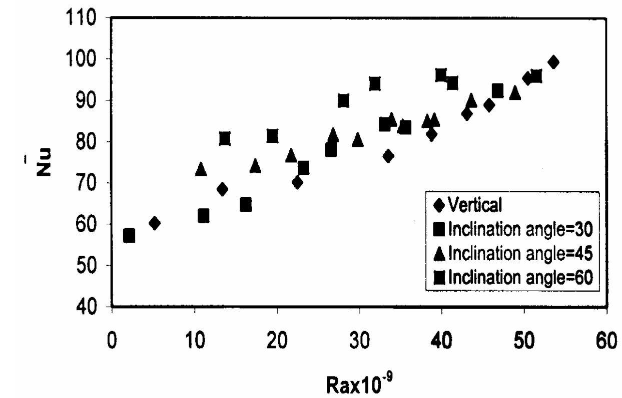
(i) For the orientation θ = 90° the length of the path of the boundary layer, which equals $D/\sin\alpha$, decreases as the inclination angle of the cylinder increases. This makes the average boundary layer thickness around the cylinder decreases with the increase of the inclination angles which leads to higher heat transfer coefficient and higher Nusselt number.



(a) Orientation angle = 0°



(b) Orientation angle = 90°



(c) Orientation angle = 180°

Fig. 7. Effect of inclination angle on average Nusselt number.

(ii) For the orientations θ = 0° and θ = 180°, the increase of the inclination angle increases the components of the buoyant force perpendicular to the upper surface of the cylinder. The increase of this force increases the rate of detachments of the boundary layer from the upper surface of the semicircular cylinder. This leads to smaller average boundary layer thickness around the cylinder and higher heat transfer coefficient and higher Nusselt number.

4.3. Empirical correlations.

Empirical correlations were developed to fit the experimental data for semicircular cylinder at different inclinations and orientations of the cylinder in fig 8. For vertical semicircular cylinder, the expression of the correlation is:

$$\overline{Nu} = 0.647\ Ra^{0.2}\ . \qquad (11)$$

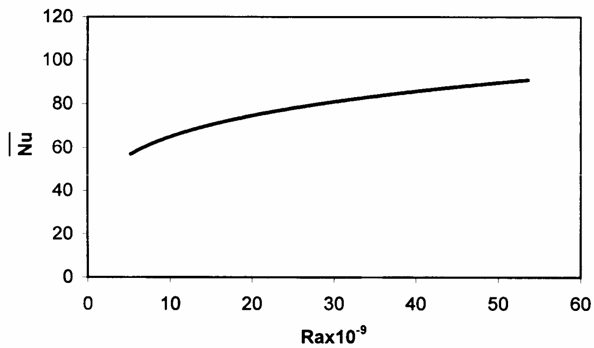
For inclined semicircular cylinder, the correlation depends on the inclination and orientation angles of the cylinder. The correlation can be put in the form:

$$\overline{Nu} = C\ Ra^{n}\ . \qquad (12)$$

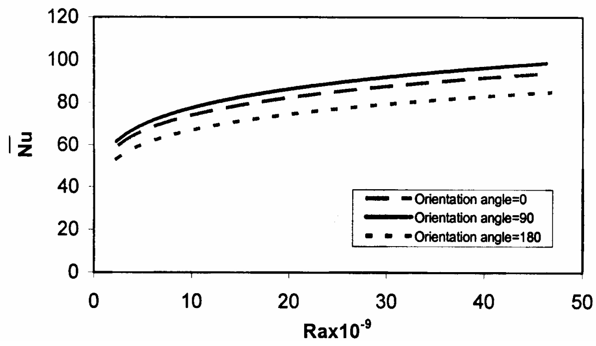
where C and n are given in table 1 for different inclination and orientation angles of the cylinder.

4.4. Comparison with literature

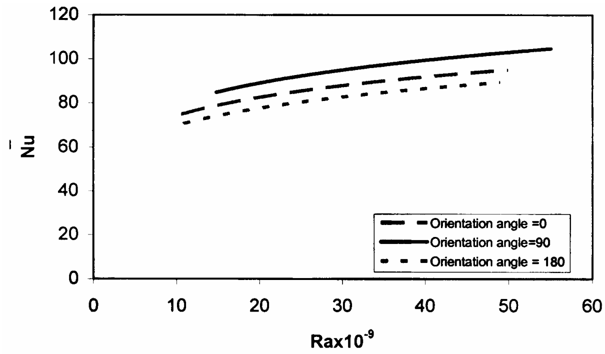
Comparison of the present work with that in the literature is not an easy work. The reason is that there is no work available in the literature for the present geometry and orientations. It was possible, however, to compare the trend of the present results with that obtained by Churchill and Chu [5] for isothermal inclined cylinder at $10^5 < Ra < 10^9$. Both works showed the increase of the average Nusselt number with increasing the inclination angle of the cylinder from the vertical. Also the trend of the present results is compared with that obtained by Nada et al. [13] for constant heat flux horizontal semicircular cylinder at $2x10^7 < Ra < 4x10^7$.



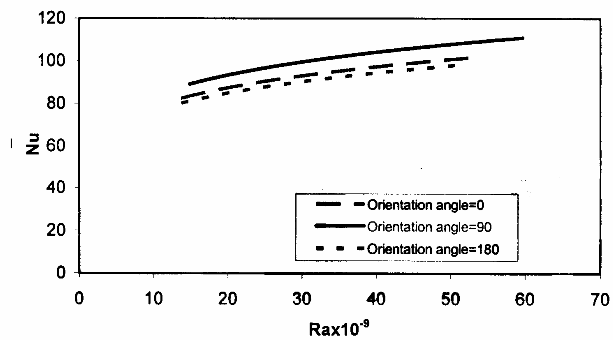
(a) Inclination angle = 0°



(b) Inclination angle = 30°



(c) Inclination angle = 45°



(d) Inclination angle = 60°

Fig. 8. Comparison between the trend lines of the correlations.

Table 1
Coefficients of eq. (10)

Inclination angle	Orientation angle					
	θ = 0°		θ = 90°		θ = 180°	
	C	n	C	N	C	N
α= 30°	2.139	0.154	2.12	0.156	1.936	0.154
α = 45°	2.1548	0.1537	2.03	0.1594	1.915	0.156
α = 60°	2.1505	0.1562	2.224	0.1576	2.1068	0.1559

The two works showed that, the Nusselt number for the orientation θ = 90° is greater than that of θ = 0° which in turn is greater than that of θ = 180°. Also the comparison showed that, for the same heat flux the average heat transfer coefficient for horizontal semicircular cylinder obtained by Nada et al. [13] for a certain orientation is higher than that obtained in the present work for α = 60° at the same orientation. This is consistent with the present results which showed the increase of the average heat transfer coefficient with increasing the inclination angle measured from the vertical.

5. Conclusions

Free convection heat transfer from vertical and inclined semicircular cylinder has been investigated experimentally at different orientations. Vertical and three different inclination angles (30, 45 and 60°) were studied at three different orientation angles of the tube (θ= 0, 90 and 180°). It was found that: (i) For any orientation of the cylinder, the Nusselt number increases with the increase of the inclination angle measured from the vertical, (ii) For any inclination of the cylinder, the Nusselt number for the orientation θ = 90° is greater than that of θ = 0° which in turn is greater than that of θ = 180°. The present work introduces correlations of Nusselt number for free convection from vertical and inclined semicircular cylinder at different orientations. The trend of the results of the present work was compared with that obtained for inclined cylinder and horizontal semicircular cylinder from literature and the trends were consistent.

Nomenclature

- D diameter of semicircular cylinder,
- dx axial distance element,
- Gr Grashof number based on constant heat flux,
- g gravitational acceleration,
- h_i local heat transfer coefficient at axial position i,
- I electric current,
- k thermal conductivity of air,
- L length of the semicircular cylinder,
- Nu_i local Nusselt number at axial position i,
- $\overline{Nu}$ average Nusselt number,
- Pr Prandtl number of air,
- q'' free convection heat transfer flux,
- q''_r radiation heat losses rate per unit area of the semicircular cylinder,
- Ra Rayleight number based on constant heat flux,
- T_f film temperature,
- T_i local surface temperature at axial position i on the cylinder surface,
- T_s average surface temperature of the semicircular cylinder,
- T_∞ ambient air temperature.
- x axial distance,
- α inclination angle of the cylinder measured from the vertical,
- β volumetric coefficient of thermal expansion of air,
- ε emissivity of the surface of the semicircular cylinder,
- μ dynamic viscosity of air,
- υ kinematics viscosity of air,
- θ orientation angle of the semicircular cylinder,
- ρ density of air, and
- σ Stefan-Boltzmann constant.

References

[1] W.H. McAdams, Heat Transmission, 3rd ed., New York, McGraw-Hill Publishing (1954).
[2] E.M. Sparrow, and J.L. Gregg, “Laminar Free Convection from a Vertical Plate”, ASME. TRANS, Vol. 78C, p. 435 (1956).

[3] C.Y. Warner, and V.S. Arpaci, “An Experimental Investigation of Turbulent Natural Convection in Air at Low Pressure for a Vertical Heated Plate”, Int. J. Heat and Mass Transfer, Vol. 11, p. 397 (1968).
[4] S.W. Churchill, and H.H.S. Chu, “Correlating Equations for Laminar and Turbulent Free Convection from a Vertical Plate”, Int. J. Heat and Mass Transfer, Vol 18, p. 1323 (1975).
[5] S.W. Churchill, Free Convection Around Immersed Bodies, In handbook of heat exchanger design, ed. G.F. Hewitt, New York: Hemisphere Publishing (1990).
[6] N.V. Suryanarayana, Engineering Heat Transfer, New York, West Publishing Company, (1995).
[7] T. Fujii, and H. Imura, “Natural Convection Heat Transfer from a Plate With Arbitrary Inclination”, Int. J. Heat and Mass Transfer, Vol. 15, p. 755 (1972).
[8] D.E. Fussey, and I.P. Warneford, “Free Convection from a Downward Facing Inclined Flat Plate”, Int. J. Heat and Mass Transfer, Vol. 21, p. 119 (1978).
[9] R.J. Goldstein, E.M. Sparrow, and D.C. Jones, “Natural Convection Mass Transfer Adjacent to Horizontal Plates”, Int. J. Heat and Mass Transfer, Vol. 16, p. 1025 (1973).
[10] J.R. Lloyd, and W.R. Moran, “Natural Convection Adjacent to Horizontal Surfaces of Various Plane Forms”, J. Heat and mass Transfer, Vol. 96, p. 443 (1974).
[11] S.W. Churchill, and H.H.S. Chu, “Correlating Equations for Laminar and Turbulent Free Convection from a Horizontal Cylinder”, Int. J. Heat and Mass Transfer, Vol. 18, p. 1049 (1975).
[12] V.Y. Morgan, The Overall Convective Heat Transfer from Smooth Circular Cylinders, In advances in heat transfer, Vol. 11, eds. T.F. Irvine, and J.P. Hartnett, New York: Academic Press. Publishing (1975).
[13] S.A. Nada, M.A. Rady, and M. Mowad, Experimental Investigation of Free Convection from a Horizontal Semicircular Cylinder at Different Orientations, Journal of Engineering Sciences, Assiut University, Egypt, Vol. 30 (3), p.731 (2002).
[14] E.M. Sparrow, and A.J. Stretton, “Natural Convection from Variously Oriented Cubs and Other Bodies of Unity Aspect Ratios”, Int. J. Heat and Mass Transfer, Vol. 22, p.747 (1985).
[15] G.D. Raithby, and K.G.T. Hollands, in W. M. Rohsenow, J.P. Hartnett and E.N Ganic, Eds. Handbook of Heat Transfer Fundamentals, Chap. 6, McGraw-Hill, New York (1985).
[16] J.H. Lienhard, A Heat Transfer Text Book, 2d ed. Englewood Cliffs, NJ: Prentice Hall Publishing (1987).
[17] A.A Busedra, and H.M. Soliman, “Experimental Investigation of Laminar Mixed Convection in an Inclined Semicircular Duct Under Buoyancy Assisted and Opposed Conditions”, Int. J. Heat and Mass Transfer, Vol 43, p. 1103 (2000).

Received December 29, 2002
Accepted April 14, 2003