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Free Convective Heat Transfer with Different Sections Lengths Placed at the Exit of a Vertical Circular Tube subjected to a Constant Heat Flux

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Abstract:

A free convective heat transfer from the inside surface of a uniformly heated vertical circular tube has been experimentally investigated under a constant wall heat flux boundary condition for laminar air flow in the ranges of Ra_L from 6.9×10^8 to 5×10^9. The effect of the different sections (restrictions) lengths placed at the exit of the heated tube on the surface temperature distribution, the local and average heat transfer coefficients were examined. The experimental apparatus consists of aluminum circular tube with 900 mm length and 30 mm inside diameter (L/D=30). The exit sections (restrictions) were included circular tubes having the same inside diameter as the heated tube but with different lengths of 600 mm (L/D=20), 900 mm (L/D=30), 1200 mm (L/D=40), 1500 mm (L/D=50), and 1800 mm (L/D=60). It was found that the surface temperature along the tube axial distance would be higher for restriction with length of 1800 mm (L/D=60) and it would be smaller for the restriction with length of 1200 mm (L/D=40). The results show that the local Nu_x and average Nusselt number $\overline{Nu}$ were higher values for the restriction with length of 1200 mm (L/D=40) and smaller values for the restriction with length of 1800 mm (L/D=60). The results were correlated with empirical equations and presented as $Log\,\overline{Nu_L}$ against $Log\,\overline{Ra_L}$ for each case investigated and a general empirical equation was proposed for all cases.

Keywords: Experimental study; Free Convective; Different sections length; Vertical Circular Tube; Constant Heat Flux.

Introduction

Free convective heat transfer has always been of particular interest among heat transfer problems. In free convection process, fluid motion is caused by density variations resulting from temperature difference between the fluid and the contacting surface. Many experimental studies have been performed during the last contacting surface. many experimental studies have been performed during the last three decades and interesting results have been presented. The free convection from cylinders or tubes of circular shapes have been receiving growing interest in the last few decades 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, **Kakac (1987)**. Heat transfer studies of free convection from circular tubes are necessary for better thermal design of industrial applications. Although some theoretical and experimental investigations have been published it is to be noted that they are far from sufficient. Experiments using different fluids and different values of length to diameter ratio in both the isothermal surface and the constant wall heat flux conditions are still needed to enable a complete investigation of the problem. Therefore, the present work was carried out in an attempt to fill a part of the existing gap and provides experimental data by experimentally investigating free convection heat transfer from the heated inside surface of a vertical tube to air at constant heat flux.

The available work on free convection from the inside surfaces of vertical tubes open at both ends with restriction at exit is limited. However, most of the available investigations are theoretical and deal with the vertical tube in special cases only. To the authors knowledge limited prior work is available on this case, which studied in the present work. **Oliver (1962)** investigated experimentally laminar flow of relatively non-viscous Newtonian liquids through a vertical jacketed tube. It was shown that better agreement was obtained when the ratio D/L is omitted from the group and further improvement results from the incorporation of the ratio L/D, all the data being adequately represented by an empirical equation. This equation becomes inaccurate when $Gz_m < \pi Nu_{am}$ and it should be mentioned that the power of L/D is only provisional

$$Nu_m\left(\frac{m_w}{m_b}\right)^{0.14} = 1.75\left(Gz_m + 5.6\times10^{-4}(Gr_m Pr_m L/D)^{0.7}\right)^{1/3} \quad (1)$$

Martin (1965) made predictions of the lower limiting conditions of free convection in the vertical open thermosyphon of circular cross-section with uniform wall temperature. The overall heat-transfer rate was independent of tube length but proportional to radius, unless the length-radius ratio is below about 1·8, in which case it depends also on temperature conditions at the closed end. The corresponding Rayleigh number was estimated for non-metallic fluids and for a liquid metal. But, **Dyer (1975)** presented a theoretical and experimental study of laminar air flow natural convective in heated vertical ducts. The temperature and velocity fields and the relationship between Nu and Ra numbers were obtained by solving the governing equations by a step-by-step numerical technique. The influence of Pr number was discussed. Experiments were conducted for Ra number between 1 to 13000. Three ducts were used of different sizes and these ducts were 19.1, 25.4, and 46.7 mm in internal diameter and were all 1220 mm in long. Comparison between experimental and theoretical studies was carried out and showed good agreement. **Kokugan and Kinoshita (1975)** performed experimental work in a heated vertical open tube consisting of heated section at constant wall temperature. Correlations between Gr and Re numbers were derived by setting up a mechanical energy balance in the tube. The following equation was proposed:

$$Gr_o = 6.3\,Re_o^{\,2} + 3.2(L_H + L_o)/(D.Re_o)\,. \quad (2)$$

Where: L_H =heated length; L_o= entrance length and subscript (o) denoted to at room temperature. The results were compared with available numerical results. **Hess and Miller (1979)** carried out experiments using a Laser Doppler Velocimeter (LDV) to measure the axial velocity of a fluid contained in a cylinder subject to constant wall heat flux on the side walls. The modified Rayleigh numbers ranged between 4.5×10^{9} to 6.4×10^{10}, which corresponds to the upper limit of the laminar regime. The variations of axial velocity with radius for different heights in the bottom and top parts of the cylinder and inside the thermal boundary layer region and with radius for different time were presented. The variation of radial position of maximum velocity and radial position of zero velocity with Rayleigh number were also depicted. Excellent agreement was obtained with the available numerical solution. **Shigeo and Adrian (1980)** studied experimentally natural convection in a vertical pipe with different end temperature with (L/D=9). The Rayleigh number was in the range $10^{8}<Ra<10^{10}$. It was concluded that the natural convection mechanism departs considerably from the pattern known in the limit Ra → 0. Specifically, the end-to-end heat transfer was affected via two thin vertical jets, the upper (warm) jet proceeding along the top of the cylinder toward the cold end and the lower (cold) jet advancing along the bottom in the opposite direction. The Nusselt number for end-to-end heat transfer was shown to vary weakly with the Rayleigh number. **Shenoy (1984)** presented a theoretical analysis of the effect of buoyancy on the heat transfer to non-newtonian power-law fluids for upward flow in vertical pipes under turbulent conditions. The equation for quantitative evaluation of the natural convection effect on the forced convection has been suggested to be applicable for upward as well as downward flow of the power-law fluids by a change in the sign of the controlling term. **Chang et al. (1986)** investigated theoretically the role of latent heat transfer in connection with the vaporization of a thin liquid film inside a vertical tube. The results were specifically presented for an air-water system under various conditions. The effects of tube length and system temperatures on the momentum, heat and mass transfer in the flow were examined. The important role that the liquid film plays under the situations of buoyancy-aiding and opposing flows was clearly demonstrated. **AL-Arabi et al. (1991)** investigated experimentally natural convection heat transfer from the inside surfaces of vertical tube to air in the ranges of $Gr_L.Pr$ from 1.44×10^{7} to 8.85×10^{8} and (L/D) ranged from 10 to 31.4. The results obtained were correlated by dimensionless groups as follows:

$$Nu_{mL} = \frac{1.11}{[\,1+0.05\,(t_{ms}-t_i\,)]}(\,Gr_{mL}\ Pr)^{0.25} \quad ..\ (3)$$

It was inferred that the effect of (L/D) on Nu_{mL} was insignificant and the entrance length was practically constant. The results were compared with the theoretical results of **Dyer (1975)** and showed a good agreement. **Abd-el-Malek and Nagwa (1991)** developed an analysis of transformation group method to study fluid flow and heat transfer characteristics for steady laminar free convection on vertical circular cylinder. The form of the surface temperature variation was derived as a linear variation with the vertical coordinate. The system of ordinary differential equations was solved numerically using a fourth-order Runge-Kutta scheme and the gradient method. The effects of the cylinder heating mode and the Prandtl number on the velocity and temperature profiles were presented. It was found that the maximum value of the vertical component of the velocity decreases with the increase of both the Prandtl number and the surface temperature. **Fukusako and Takahashi (1991)** investigated the influence of density inversions and free convection heat transfer of air-water layers in a vertical tube with uniformly decreased wall temperature. Holographic interferometry was adopted to determine the time-dependent temperature distribution in the tube. The temperature and the flow patterns were markedly influenced by the cooling rate of the tube. The heat transfer characteristics along the tube wall were also determined. **Yan and Lin (1991)** performed combined theoretical and experimental study to investigate natural convection in vertical pipe flows at high Rayleigh number. The wall conduction effects and thermal property variations of the fluid and pipe wall were also considered. The predicted and measured distributions of wall temperature and Nusselt number were in good agreement. The empirical correlations for the induced flow rate and average Nusselt number were proposed. **Vinokurov et al. (1993)** carried out an experimental investigation of unsteady-state free convection in a vertical cylindrical channel for the case of non-uniform distribution of heat flux along a channel at a constant wall temperature. The averaged temperature field in a gas was investigated on a Mach-Zender interferometer. Hydrodynamic structures were investigated by the smoke visualization technique. The longitudinal and lateral Rayleigh numbers were varied from 0 to 4×10^9 and from 0.8×10^4 to1.2×10^5, respectively. The air, carbon dioxide and helium were used in this study as working fluids. **Yissu (1995)** examined numerically and experimentally laminar natural convection in vertical tubes with one end open to a large reservoir, to predict the flow behavior and the heat transfer rates. In the numerical study, a semi-implicit, time-marching, finite-volume solution procedure was adopted to solve the governing equations. The experimental work involved the use of a Mach-Zehnder interferometer to

examine the temperature field for a modified rectangular open thermosyphon through the

interpretation of fringe patterns. The Nusselt numbers were determined from the interferometer results and compared with numerical results. It was found that the heat transfer rates through the tube wall to be strong functions of the tube radius, and approached an asymptotic limit as the tube radius was increased. **Kuan-Tzong Lee (2000)** presented a closed form solution for the fully developed laminar natural convection heat and mass transfer in a vertical partially heated circular duct. Thermal boundary conditions of uniform wall temperature/uniform wall concentration (UWT/UWC) and uniform heat flux /uniform mass flux (UHF/UMF) were considered. **Wojciech (2000)** presented experimental and numerical studies of natural convection in a vertical tube placed between two isothermal walls of different temperature. Two experimental set-ups were built for visualizing the flow and to measure the temperature around the tube and walls confined in the close cavity. An FEM computer code was applied for analyzing the influence of various parameters on the flow structure and heat transfer. The measurements were taken for Rayleigh number: for slot $Ra_s=2\times10^7–1\times10^9$; for tube $Ra_r=6\times10^2$ - 9×10^2. It was found that the intensity decreased and increased unexpectedly and the explanation can be given after visualization. It was observed that the hot air near the heated wall aspirated the layers of hot air from the vertical tube. The uplift pressure and the tendency of the system were reached the balance because the air layers of the similar temperature and density were merged. However, this effect (similar to the movement of the hot air in a chimney) also found to be dependent on the intensity of the heat transfer. **He et al. (2004)** investigated numerically natural convection

heat transfer and fluid flow in a vertical cylindrical envelope with constant but different temperatures of the two end surfaces and an adiabatic lateral wall. The simulation was conducted for two end wall temperature differences: ΔT_w=10 and 220 K. For the cases of ΔT_w=10 K, it was found that the variation patterns of Nu_L versus Ra_L within the range of L/D=3-10 were in good consistency with the available experimental and theoretical results. For the case of large temperature difference (ΔT_w=220 K) the natural convection in the enclosure was quite strong in that the convective heat transfer rate was about two-orders larger than that of pure heat conduction which occurs when the cold end is placed down. The numerical simulation also revealed that the ratio of the axial length, L, to the diameter, D, has effect on the average heat transfer rate of the envelope under the same other conditions. It was concluded that within the range of L/D=1–9, the increase in L/D leads to the decrease in heat transfer rate. **Popiel and Wojtkowiak (2004)** presented an experimental investigation on average natural convection heat transfer from the isothermal vertical surfaces of a short and slender square cylinder to air obtained with a lumped capacitance method. A validation of experiments was performed with the short circular cylinder and showed very good agreement with the formula of **(McAdams, 1974)** which is recommended for a flat isothermal plate. **Moawed (2005)** studied experimentally natural convection from uniformly heated helicoidal pipes oriented vertically and horizontally. Four helicoidal pipes of different parameters were presented. The effects of pitch to pipe diameter ratio, coil diameter to pipe diameter ratio and length to pipe diameter ratio on the average heat transfer coefficient were found. The experiments covered a range of Rayleigh number based on tube diameter from 1.5×10^3 to 1.1×10^5. The results showed that the overall average Nusselt number, Nu_m, increases with the increase in pitch to pipe diameter ratio, coil diameter to pipe diameter ratio and length to pipe diameter ratio.

It is clear from the above literature review that there are limited studies which deal with the effect of the restrictions placed at the exit of a vertical circular tube on the heat transfer from the inside surface with different sections (restrictions) lengths. Thus, the purpose of the present study was to provide experimental data on free convective heat transfer from open ended vertical circular tube with a constant heat flux and with different sections (restriction) lengths and to propose a general empirical equation for this problem.

Experimental apparatus and procedure

The experimental apparatus used in the present work consists of a circular tube (test section) mounted on a specially constructed wooden frame. The frame has the capability of rotating the test section around its horizontal axis and changing its inclination angle. The heated section proceeded with circular sections (restrictions) with different lengths, as well as, different Grashof number as shown schematically in **Fig.1a**, to investigate the free convection heat transfer in a vertical circular tube opens at both ends.

The heated tube (3) is made from aluminum tube of 900 mm length, 30 mm inside diameter (L/D=30) with a thickness of 5 mm provided with changeable restriction tubes (1) with five different lengths, particulars of which are: cylindrical tubes with lengths of 600 mm ($L/D_{rest.}$=20), 900 mm ($L/D_{rest.}$=30), 1200 mm ($L/D_{rest.}$=40), 1500 mm ($L/D_{rest.}$=50), and 1800 mm ($L/D_{rest.}$=60) placed at the exit of the heated section as shown in **Fig. 1a**. The air enters from the atmosphere through the circular (restriction) tube into the heated section and then the heated air was exhausted to the atmosphere. The teflon connection pieces:

represents a part of the test section inlet (4) and another teflon piece represents the test section exit (5). The restriction tubes were connected with the heated tube by teflon connection piece (4) bored with the same inside diameter of the heated tube and the restriction tube as shown in **Fig. 1a**. The teflon was chosen because its low thermal conductivity in order to reduce the test section ends losses. The outer surface of the tube was covered with an electric insulating tape on which nickel-chrome wire (3) of 0.4 mm is electrically isolated by ceramic beads, uniformly wounded along the tube as a coil in order to give uniform heat flux as shown in **Fig. 1b**. The outside of the test section was then thermally insulated by an asbestos layer (4) with 20 mm thickness, and with 18 mm thickness of fiberglass layer (5). Two pairs of thermocouples (6) were installed in the asbestos layer between the heater and the insulation at three stations along the heated section as shown in **Fig. 1b** in order to perform the heat loss calculation through the test section lagging. The heat losses from the ends of the test section could be evaluated by inserting two thermocouples in each teflon piece. By knowing the distance between these thermocouples and the thermal conductivity of the teflon, the end losses could be calculated. The thermocouples of each pair were fixed on the same radial line. The input power to the heater was adjusted so that at steady state, the readings of the thermocouples (6) of each pair become practically the same. Thirty five alumel-chromel (type K) thermocouples (2) of 0.2 mm diameter were soldered in slots milled in the axial direction to measure the surface temperature of the test tube as shown in **Fig. 1b**. The measuring junctions were permanently secured in the holes by sufficient amount of high temperature application defcon adhesive (4) as shown in **Fig. 1c**. All thermocouples were calibrated using the melting points of ice made from distilled water as reference point and the boiling points of several pure chemical substances. The calibration of thermocouples showed that they were accurate to within ± 0.2 °C. The inlet bulk air temperature was measured by one thermocouple placed at the beginning of the restriction tube, while the outlet bulk air temperature was measured by two thermocouples located at the test section exit 'mixing chamber' (7) as shown in **Fig. 1a**. The local bulk air temperature was calculated by fitting straight line-interpolation between the measured inlet and outlet bulk air temperatures since the wall heat flux boundary condition is applied. The choice of the linear distribution of the bulk air temperature is attributed to the following reasons: for constant wall heat flux (q) boundary condition, the bulk temperature gradient is calculated from:

$$\frac{dT_b}{dx} = \frac{q.p}{\dot{m}Cp} = \frac{p}{\dot{m}\,Cp} h(T_s - T_b) \ldots\ldots.. \qquad (4)$$

Where p is the surface perimeter $= \pi D$ for circular tube. From the above equation the axial variation of T_b may be determined. If $T_s > T_b$ heat is transferred to the fluid and T_b increases with x, if $T_s < T_b$ the opposite is true. For constant heat flux (q) it follows the right hand side of Eq. (4) is a constant independent of the distance (x), hence,

$$\frac{dT_b}{dx} = \frac{q.p}{\dot{m}Cp} \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \qquad (5)$$

by integrating from x=0, it follows that

$$T_b(x) = T_{b,i} + \frac{q.p}{\dot{m}Cp} x \ldots\ldots\ldots\ldots\ldots\ldots. \qquad (6)$$

Accordingly, the bulk temperature varies linearly with the distance (x) along the tube. Moreover, from

$$q = h\,(T_s - T_b) \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots..(7)$$

The temperature difference (T_s-T_b) varies with the distance (x). The difference is initially small (due to the large value of the heat transfer coefficient at the tube entrance) but increases with increasing the distance (x) due to the decrease in heat transfer coefficient that occurs as the thermal boundary layer develops as has been reported by **Incropera and DeWitt (2003).**

The readings of all thermocouples were taken by a precalibarted digital temperature recorder capable of reading 0.01 °C via a multi-switch. The apparatus was mounted in a closed room with plastic transparent shields (2) as shown in **Fig. 1a** to prevent currents of air, and the measuring instruments were mounted outside of this room. The input electric power to the heater was controlled and changed by the AC variac at each experiment and measured by a digital wattmeter with a resolution of 0.01 W. The steady state condition for each run was achieved after 4 approximately hours. The steady state was considered to be achieved when the temperature reading of each thermocouple did not change by more than 0.5 °C within 20 minutes. When the steady state condition was established, the readings of all thermocouples, the input power and the inlet and outlet bulk temperatures were recorded.

Experimental Uncertainty

Generally the accuracy of experimental results depends upon the accuracy of the individual measuring instruments and the manufacturing accuracy of the circular tube. The accuracy of an instrument is also limited by its minimum division (its sensitivity). In the present work, the uncertainties in heat transfer coefficient (Nusselt number) and Rayleigh number were estimated following the differential approximation method reported by **Holman (2001)**. For a typical experiment, the total uncertainty in measuring the heater input power, temperature difference (T_s-T_a), the heat transfer rate and the circular tube surface area were 0.38%, 0.48%, 2.6, and 1.3% respectively. These were combined to give a maximum error of 2.43% in heat transfer coefficient (Nusselt number) and maximum error of 2.36% in Rayleigh number.

Data Reduction

In the present work the following steps were used to analyze the natural convection heat transfer process for air flow in a vertical circular tube when its surface was subjected to a constant wall heat flux boundary condition.

The total input power supplied to the heated tube can be calculated:

$$Q_t = I^2 \times R \quad \text{.................................}(8)$$

The convection heat transferred from the heated tube surface:

$$Q_{conv.} = Q_t - Q_{cond.} \quad \text{.........................} \ (9)$$

Where: $Q_{cond.}$ is the total conduction heat losses (lagging and ends losses) and its calculated from $\left(Q_{cond.} = DT / R_{th}\right)$

Where: $DT = \overline{T}_{fiberglass\ layer} - \overline{T}_{asbestos\ layer}$ and R_{th} is the thermal resistance of the insulations

$$R_{th} = \frac{ln(r_o / r_i)}{2p\, k_{insulations} L}$$

Where: r_o is the outer radius of the heated tube and r_i is the inner radius of the heated tube. It was found that the conduction heat losses from the heated section are approximately about 4% of the total input power.

The convection heat flux can be represented by:

$$q_{conv.} = \frac{Q_{conv.}}{A_s} \quad \text{……………………..} \quad (10)$$

Where: $A_s = \pi \times D \times L$

The convection heat flux, which is used to calculate the local and average heat transfer coefficient as follows:

$$h_x = \frac{q_{conv.}}{T_{sx} - T_{bx}} \quad \text{……………..……} \quad (11)$$

Where: T_{sx} is the local surface temperature, and T_{bx} is the local bulk air temperature.

All the air properties were evaluated at the mean film temperature as reported in **Cengel (2004)**.

$$T_{fx} = \frac{T_{sx} + T_{bx}}{2} \quad \text{……………… ..} \quad (12)$$

Where: T_{fx} is the local mean film air temperature.

The local Nusselt number (Nu_L) can be determined as:

$$Nu_L = \frac{h_x . L}{k} \quad \text{……………………..} \quad (13)$$

The average values of Nusselt number $(\overline{Nu_L})$ can be calculated based on the average heat transfer coefficient as follows:

$$\overline{h_L} = \frac{1}{L} \int_{x=0}^{x=L} h_x \, dx \quad \text{…………………. …..} \quad (14)$$

$$\overline{Nu}_L = \frac{\overline{h_L} . L}{k} \quad \text{………………………} \quad (15)$$

The average values of the surface temperature, bulk air temperature and mean film temperature can be evaluated as follows:

$$\overline{T_s} = \frac{1}{L} \int_{x=0}^{x=L} T_{sx} \, dx \quad \text{………………….. ….} \quad (16)$$

$$\overline{T_a} = \frac{1}{L} \int_{x=0}^{x=L} T_{bx} \, dx \quad \text{…….. ………} \quad (17)$$

$$\overline{T_f} = \frac{\overline{T_s} + \overline{T_a}}{2} \quad \text{…………………….…..…} (18)$$

.

The Grashof and the Rayleigh numbers can be determined as follows:

$$\overline{Gr}_L = \frac{g \, b \, L^3 (\overline{T_s} - \overline{T_a})}{n^2} \quad \text{……….…….} (19)$$

$$\overline{Ra}_L = \overline{Gr}_L \times Pr \quad \text{…………………………} \quad (20)$$

Where: $b = 1/(273 + \overline{T_f})$, All the air physical properties (r, m, n and k) were evaluated at the average mean film temperature ($\overline{T_f}$), but it was observed in most of the previous work investigations that the physical properties were taken at the mean film temperature which is based on ambient temperature at tube entrance and given by $[T_{mf} = (T_{ms} + T_i)/2]$.

Results and discussion

Free convection of air was experimentally studied in a vertical circular tube of different sections (restrictions) lengths placed at the exit of the heated tube. The heated tube was subjected to constant wall heat flux boundary conditions. The effects of the restriction length and Ra number on the heat transfer results were discussed in this section. The results presented in this paper include the surface temperature distribution of the heated tube, local Nu number and average Nu number. The present experimental data covered a total of 40 test runs for five restrictions with different lengths of 600 mm (L/D=20), 900 mm (L/D=30), 1200 mm (L/D=40), 1500 mm (L/D=50), and 1800 mm (L/D=60) with the range of heat flux from 249 W/m^2 to 1260 W/m^2.

Surface temperature

The distribution of the surface temperature along the tube axial distance may be affected by many variables such as the heat flux, and the restriction length. The surface temperature distribution for selected runs is plotted and shown in **Figs.2-6**. The distribution of the surface temperature (T_s) with tube axial distance for different heat fluxes and for all restriction lengths have the same general shape. The surface temperature distribution exhibits the following trend: the surface temperature gradually increases with the axial distance until a certain limit to reach a maximum value at approximately (X/D=5.84) beyond which it begins to decrease. This phenomenon, which can be explained as follows: at the entrance to the tube the thickness of the thermal boundary layer is zero. Then, it gradually increases until the boundary layer fills the tube. From the entrance of tube to maximum point the heat transfer gradually decreases and (T_s) gradually increases due to the laminarization effect in the near wall region (buoyancy effect) and due to the upstream axial conduction in the solid walls preheating the air in the restriction section and due to tube end losses. Beyond the maximum point one would surmise that the surface temperature decreases along the axial distance. However, as the air is heated along the tube, its physical properties gradually change with the increased temperature. The thermal conductivity increases causing less resistance to the flow of heat and the viscosity increases causing radial flow of the hotter layers of air nearer to the surface to the tube center. A gradual increase of the local heat transfer beyond the maximum point must then be appeared. For constant wall heat flux this can only take place if the local differences between the bulk air temperature and the surface temperature decreases as shown in the distribution of (T_s-x). **Fig. 2** shows the distribution of the surface temperature along the tube for different heat fluxes, for restriction with length of 600 mm ($L/D_{rest.}$=20). This figure reveals that the surface temperature increases at tube entrance to reach a maximum value after which the surface temperature decreases. This can also be attributed to the developing of the thermal boundary layer faster due to buoyancy effect as the heat flux increases, and as explained previously. **Fig. 3** is similar to **Fig. 2** but pertains to a restriction with length of 900 mm ($L/D_{rest.}$=30). The curves in the two figures show similar trend, but the surface temperature values in **Fig. 3** were higher than that observed in **Fig. 2** due to the length of restriction tube. **Figs. 4-6** were similar in trends to **Figs. 2-3** but pertains to restrictions with length of 1200 mm ($L/D_{rest.}$=40) in **Fig. 4**, 1500 mm ($L/D_{rest.}$=50) in **Fig. 5** and 1800 mm ($L/D_{rest.}$=60) in **Fig. 6** respectively. **Fig. 7** shows the effect of variation of restrictions lengths on the tube surface temperature for high heat flux 1260 W/m^2. It is obvious from that the surface temperature increases as the restriction length increases, as the heat flux is kept constant except the restriction section with length of 1200 mm ($L/D_{rest.}$=40) because it has the lowest surface temperature than other restriction sections. It is necessary to mention that the friction between the inside surface of the restriction length and the air flowing through it caused the temperature at entrance of the heated tube to be higher than the ambient temperature. It was also apparent from **Fig.7** that the lower values of the surface temperature take place in ($L/D_{rest.}$=40) and the higher values occur in

($L/D_{rest.}$=60) since the mass flow rate through the heated tube is the main parameter influencing the heat transfer results, so for the restriction tube with ($L/D_{rest.}$=40) gives smallest flow resistance and maximal mass flow rate and finally lower surface temperature.

local nusselt number (Nu_x)

For free convection from a uniformly heated surface of length (L) exposed directly to the atmosphere, the average heat transfer coefficient for the whole length is calculated from:

$$\overline{h_L} = \frac{1}{L} \int_{x=0}^{x=L} h_x \, dx \quad \text{.........................(21)}$$

Where: $h_x = \dfrac{q}{DT} = \dfrac{q}{T_{sx} - T_{bx}}$ .... (22)

(ΔT) in the above equation was taken as the difference between that local surface temperature (T_{sx}) and the air temperature far away the effect of the surface. Most of the previous workers mentioned in introduction section have calculated the heat transfer coefficient based on the temperature difference between the surface temperature and the fluid temperature at the entrance (T_i) [i.e. (ΔT) = (T_{sx}-T_{bi})]. In the present work, since the heat transfer surface is not exposed to the atmosphere (because the flow is confined). So that, the heat is transferred from the hot surface of the tube to the air flowing in it. Thus, $(\Delta T)_x$ cannot be taken equal to (T_s-T_i). It should be taken as (T_{sx}-T_{bx}) where (T_{bx}) is the local bulk air temperature in the cylinder.

The distribution of the local Nusselt number (Nu_x) with the dimensionless axial distance (X/D) is plotted for selected runs and shown in **Figs. 8-14**.

Figs. 8-12 show the effect of the heat flux variation on the Nu_x distribution for the five restrictions lengths under consideration in the present work. It is clear from these figures that at the higher heat flux, the results of Nu_x were higher than the results of lower heat flux. This may be attributed to the secondary flow effect that increases as the heat flux increases leading to higher heat transfer coefficient. Therefore, as the heat flux increases, the fluid near the wall becomes hotter and lighter than the bulk fluid in the core. As a consequence, two upward currents flow along the sides walls, and by continuity, the fluid near the tube center flows downstream. **Figs. 13 & 14** show the effect of the restriction section length variation on the Nu_x distribution with (X/D), for low heat flux 249 W/m^2 and for high heat flux 1260 W/m^2. For constant heat flux, the Nu_x values give higher results for the restriction tube with length of 1200 mm ($L/D_{rest.}$=40) and the lower values occur in the restriction tube with length of 600 mm ($L/D_{rest.}$=20). This situation reveals that in ($L/D_{rest.}$=40) the flow will be faster and then it makes heat transfer enhancement rather than other restriction tubes as a result of density increasing and the buoyancy force decreasing which lead to change in the temperature gradient as well as change the volume of upward gases because of the flow area decreases with the increasing of the velocity of upward gases. In addition, since the velocity profile is fully developed at the entrance of the heated section, so the restriction tube will become as a resistance on the air flow and as the length to diameter ratio (L/D) of the restriction tube was higher, the flow resistance will be higher, so that the surface temperature will be higher, and this leads to lower values of Nu_x when the restriction tube length increases. Finally, the mass flow rate through heated tube which is the main parameter influencing the heat transfer results, causes the maximum heat transfer will be occurred in ($L/D_{rest.}$=40), which gives smallest flow resistance, maximal mass flow rate and higher heat transfer results. The presented results have shown qualitatively the same trend and behaviors as observed by **He et al. (2004)**.

average nusselt number $(\overline{Nu})$

The distribution of the average Nusselt number with the dimensionless axial distance (X/D) is depicted for selected runs in **Figs. 15-16** which show the effect of the heat flux variation on the $(\overline{Nu})$ for restrictions with lengths of ($L/D_{rest.}$=20) and ($L/D_{rest.}$=40) respectively. The $(\overline{Nu})$ variation for other restriction lengths has similar trend as mentioned for ($L/D_{rest.}$=20) and ($L/D_{rest.}$=40).

average heat transfer correlation

The general correlation obtained from dimensional analysis for heat transfer by free convection available in **Incropera and Dewitt (2003)**:

$$\overline{Nu} = f_1\,(\overline{Gr, Pr})^n \quad \text{(23)}$$

In the case of heat transfer from the inside surface of a vertical tube one expects that there is an effect of both length and diameter. For similarity with flat surface (which a cylinder of infinite diameter) the characteristic linear dimension in Nu and Gr numbers may be taken as the tube length (L) since the heated tube is vertically oriented. Then equation (19) becomes:

$$\overline{Nu_L} = f_2\,(\overline{Gr_L, Pr})^n \quad \text{(24)}$$

The following correlations were obtained from the present work for each restriction length and a general correlation for all investigated cases was proposed and shown in **Fig. 17**:

$$\overline{Nu_L} = 0.88\,(\overline{Ra_L})^{0.23}$$ For restriction with ($L/D_{rest.}$=20) (25)

$$\overline{Nu_L} = 1.024(\overline{Ra_L})^{0.23}$$ For restriction with ($L/D_{rest.}$=30) (26)

$$\overline{Nu_L} = 1.068\,(\overline{Ra_L})^{0.23}$$ For restriction with ($L/D_{rest.}$=40) (27)

$$\overline{Nu_L} = 1.036\,(\overline{Ra_L})^{0.23}$$ For restriction with ($L/D_{rest.}$=50) (28)

$$\overline{Nu_L} = 1.042\,(\overline{Ra_L})^{0.23}$$ For restriction with ($L/D_{rest.}$=60) (29)

$$\overline{Nu_L} = 1.263(\overline{Ra_L})^{0.23}$$ For all restriction lengths (30)

A comparison was made between the present case and with the vertical tube open at both ends without any restriction and with the case of putting the restriction tube at the entry of the heated tube as reported by **Salman and Mohammed (2005)** and this comparison was shown in **Fig. 17**, the correlation obtained for the normal case as reported in **McAdams (1954)** and it has this form:

$$\overline{Nu_L} = 0.59\,(\overline{Ra_L})^{0.25} \quad \text{(31)}$$

From the comparison, it is apparent that the restriction length and position have a significant effect on the heat transfer results.

Conclusions

Laminar free convection heat transfer from the inside surface of a uniformly heated vertical circular cylinder with different restriction lengths placed at the exit of the heat tube was experimentally performed. The following conclusions can be drawn from this work as:

1. The length of restriction tube provides a measure of the severity. An important finding of this study was that if the restriction exceeds a certain size, laminar upward flow throughout the whole of the heated part of the tube is tends to decrease the heat transfer results. This occurs if the ratio of the unheated length to the diameter of the tube (L/D) exceeds 40.
2. It was observed that the hot air near the heated wall aspirated the layers of hot air from the vertical tube. The uplift pressure and the tendency of the system were reached a balance because the air layers of the similar temperature and density were merged. However, this effect (similar to the movement of the hot air in a chimney) also found to be dependent on the intensity of the heat transfer.
3. The experimental results have revealed that the ratio of the axial length (L) to the diameter (D) affects the average heat transfer rate under the same conditions. Within the range of $L/D_{rest.}$=20-60, the increase in L/D leads to decrease the heat transfer rate except the case of $L/D_{rest.}$=40.
4. For the same heat flux, the surface temperature values for restriction with ($L/D_{rest.}$=40) were lower than that for other restriction lengths.
5. For the same heat flux, the Nu_x values for ($L/D_{rest.}$=40) were higher than that for other restriction lengths.
6. Empirical equations in the form of $Log\,\overline{Nu_L}$ versus $Log\,\overline{Ra_L}$ were obtained for each restriction length, Eqs. 25-29, and a general correlation for all cases restriction lengths was proposed (Eq. 30).
7. The comparison with the previous work shows that the restriction length and position has a significant effect on the heat transfer results.

Nomnclature			
A_s	Tube surface area, (m^2)	**Dimensionless Group**	
C_p	Specific heat at constant pressure, (kJ/kg. °C)	Gr:	Grashof number, $g\, b\, L^3(T_s-T_a)/n^2$
D	Tube diameter, (m)	Nu:	Nusselt number, $h.L / k$
g	Gravitational acceleration, (m/s^2)	Pr:	Prandtl number, $m\,.\,Cp/k$
h	Heat transfer coefficient, $(W/m^2.°C)$	Ra:	Rayleigh number, $Gr.Pr$
I	Heater current, (ampere)	X/D:	Dimensionless axial distance
K	Thermal conductivity, (W/m. °C)		
L	Tube length, (m)	**Subscript**	
$Q_{cond.}$	Conduction heat loss, (W)	a	air
$q_{conv.}$	Convection heat flux, (W/m^2)	b	bulk
$Q_{conv.}$	Convection heat loss, (W)	f	film
Q_t	Total heat input, (W)	i	inlet
T V	Temperature, (°C) Heater voltage, (volt)	L m rest.	based on tube length mean restriction
		s	surface
Greek		t	total
b	Thermal expansion coefficient, (1/K)	w	wall
m	Dynamic viscosity, (kg/m.s)	x	local
n r	Kinematic viscosity, (m^2/s) Air density, (kg/m^3).	**Superscript**	
		——	average

1. Restriction tube
2. Shields
3. Heated tube
4. teflon connection piece
5. Exit teflon piece
6. Thermocouples
7. Mixing chamber

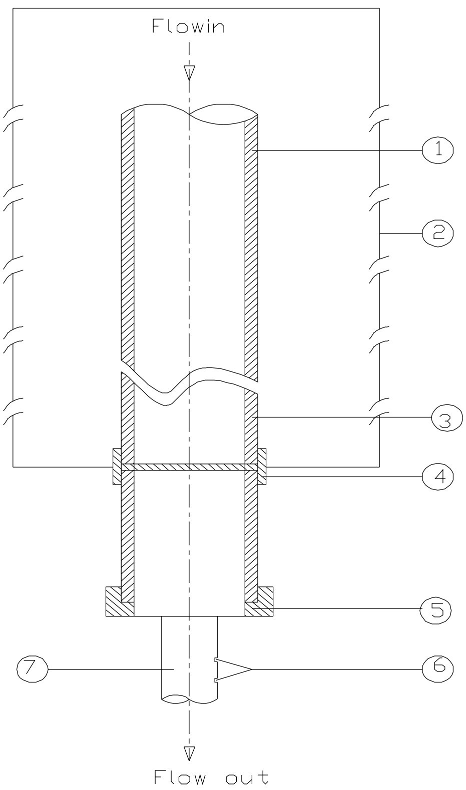

Fig. 1a The layout of experimental apparatus

1. Heated tube
2. Surface thermocouple
3. Electric heater
4. Asbestos layer
5. Fiberglass layer
6. Pair of thermocouples

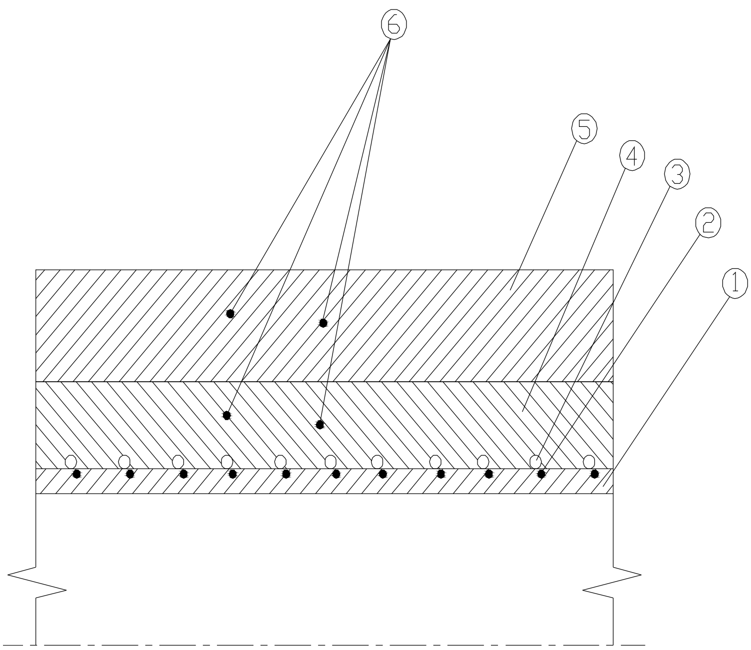

Fig. 1b The heating arrangement

1. Heated tube
2. Restriction tube
3. Thermocouples
4. Defcon adhesive

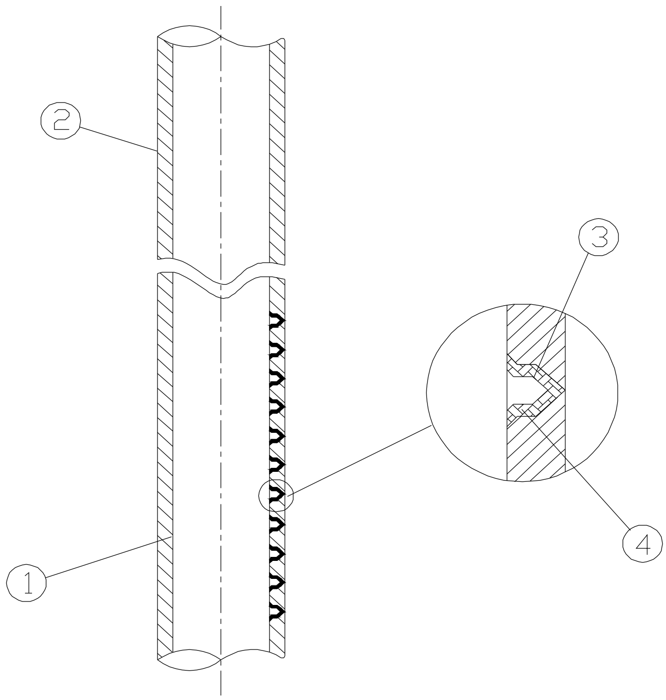

Fig. 1c The thermocouple locations along the heated tube

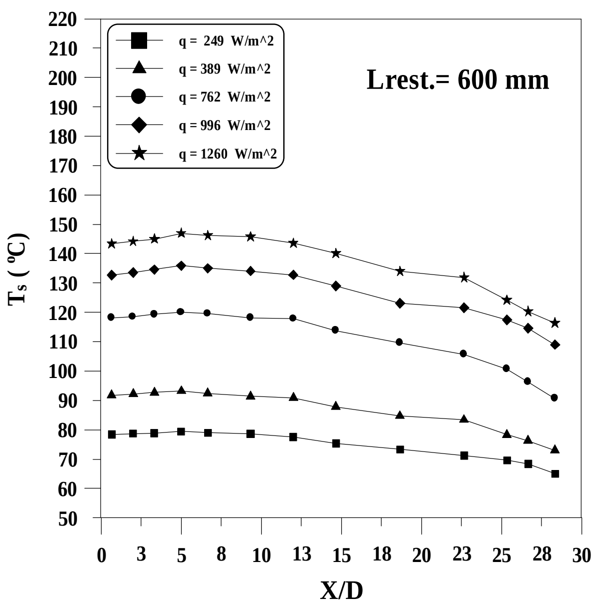

Fig.2 Variation of the surface temperature with the axial distance for ($L/D_{rest.}$=20)

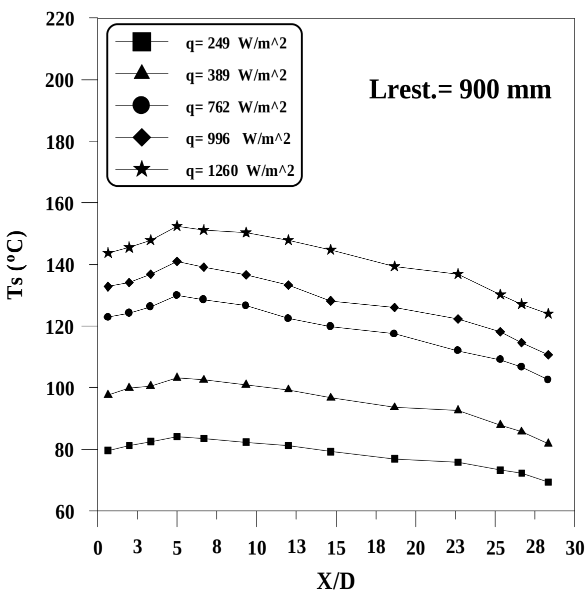

Fig. 3 Variation of the surface temperature with the axial distance for ($L/D_{rest.}$=30)

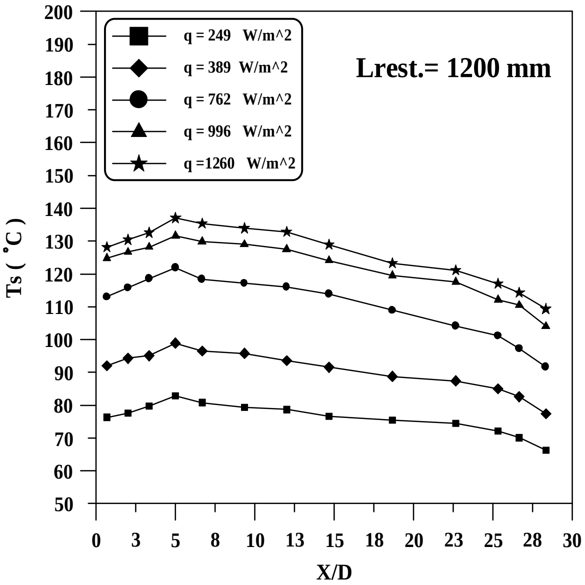

Fig.4 Variation of the surface temperature with the axial distance for ($L/D_{rest.}$=40)

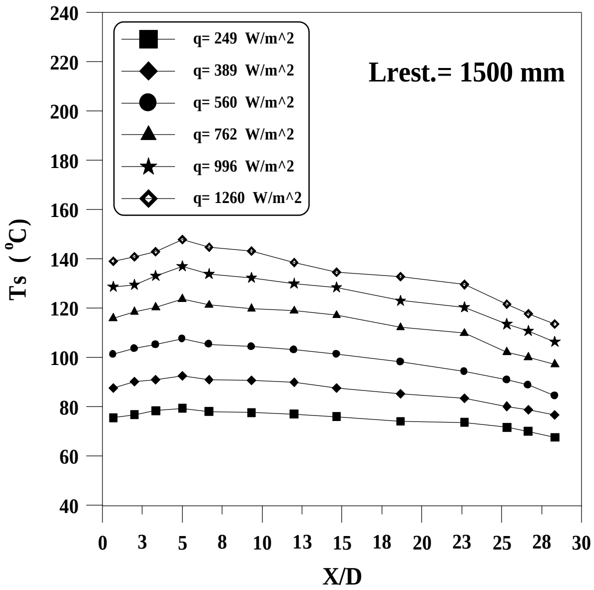

Fig.5 Variation of the surface temperature with the axial distance for ($L/D_{rest.}$=50)

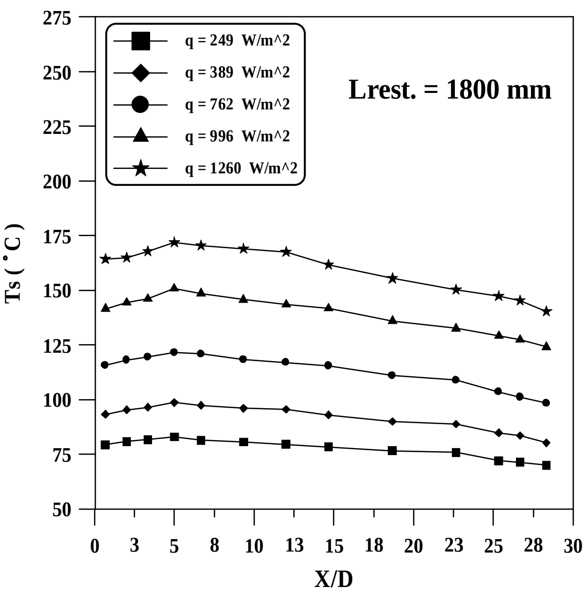

Fig. 6 Variation of the surface temperature with the axial distance for ($L/D_{rest.}$=60)

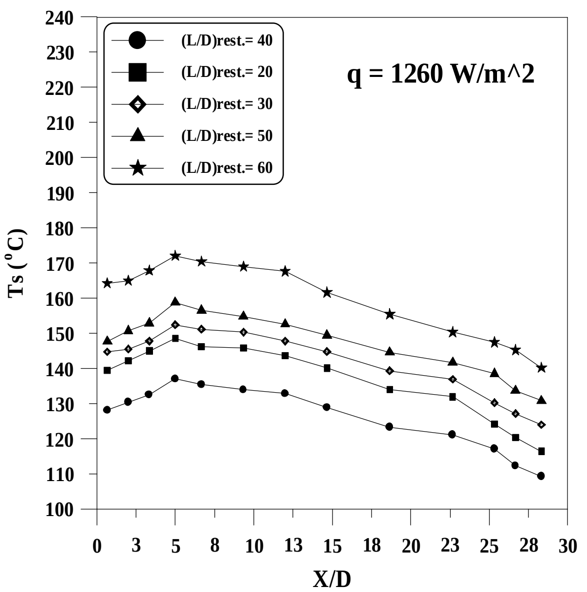

Fig. 7 Variation of the surface temperature with the axial distance for different restriction lengths

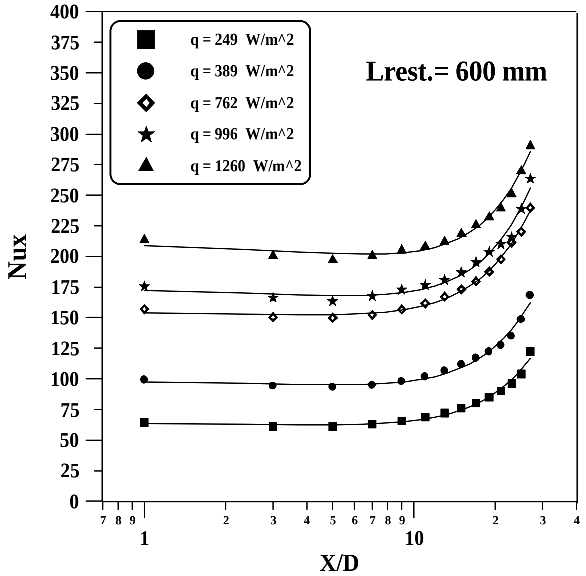

Fig. 8 Variation of the local Nusselt number with the axial distance for ($L/D_{rest.}$ =20)

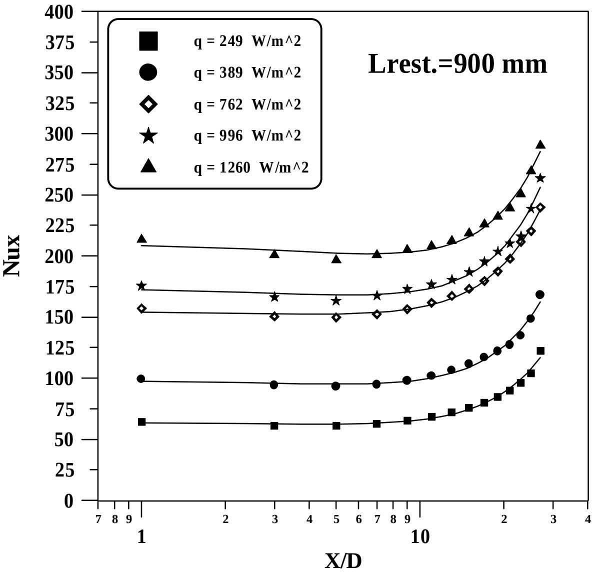

Fig. 9 Variation of the local Nusselt number with the axial distance for ($L/D_{rest.}$ =30)

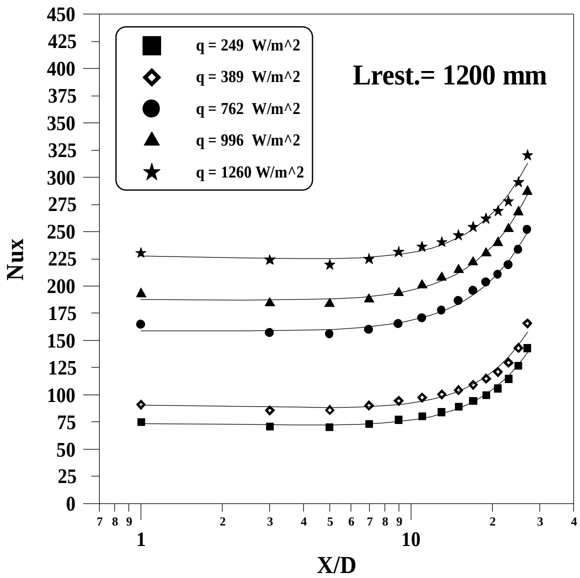

Fig. 10 Variation of the local Nusselt number with the axial distance for ($L/D_{rest.}$ =40)

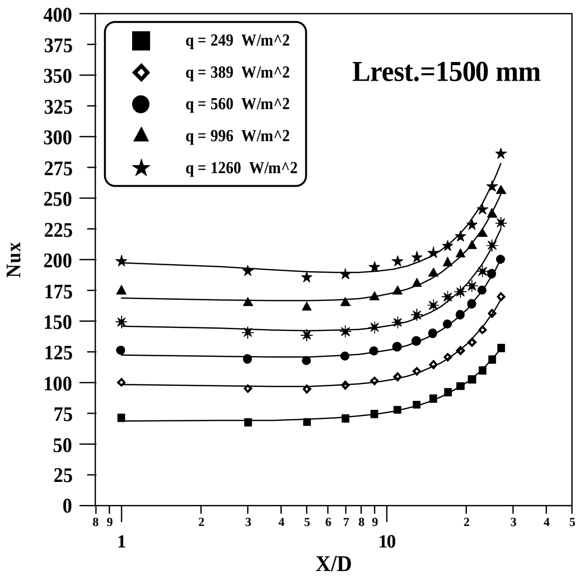

Fig. 11 Variation of the local Nusselt number with the axial distance for ($L/D_{rest.}$ =50)

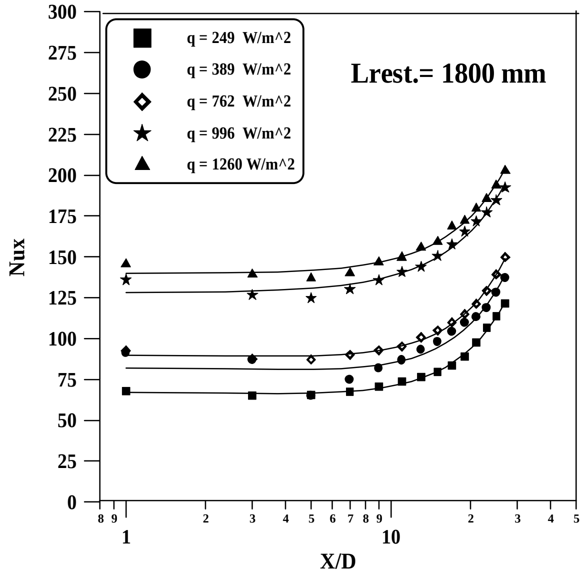

Fig. 12 Variation of the local Nusselt number with the axial distance for ($L/D_{rest.}$ =60)

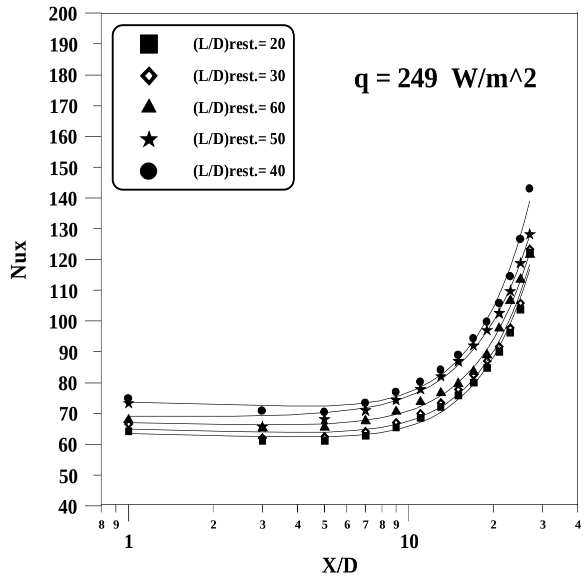

Fig. 13 Variation of the local Nusselt number with the axial distance for different restriction lengths

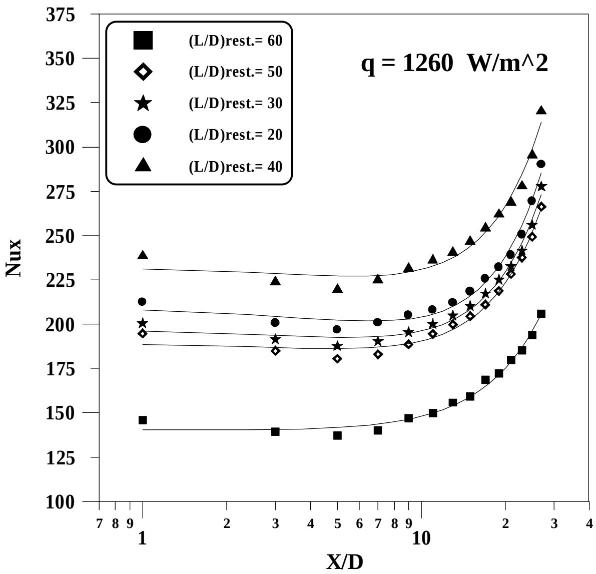

Fig. 14 Variation of the local Nusselt number with the axial distance for different restriction lengths

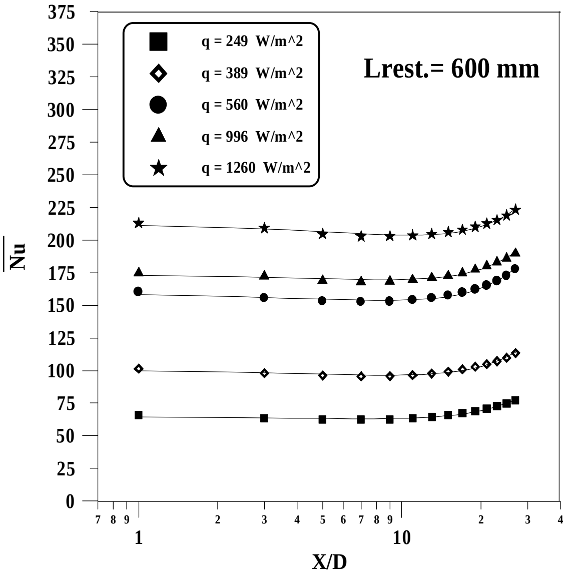

Fig. 15 Variation of the average Nusselt number with the axial distance for ($L/D_{rest.}$ =20)

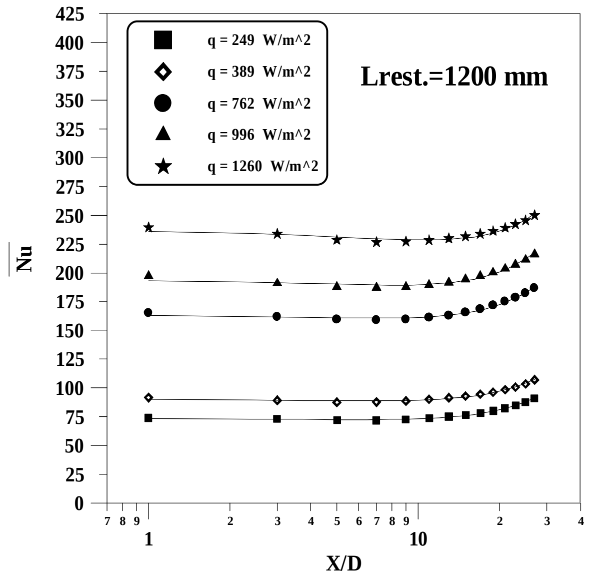

Fig. 16 Variation of the average Nusselt number with the axial distance for ($L/D_{rest.}$ =40)

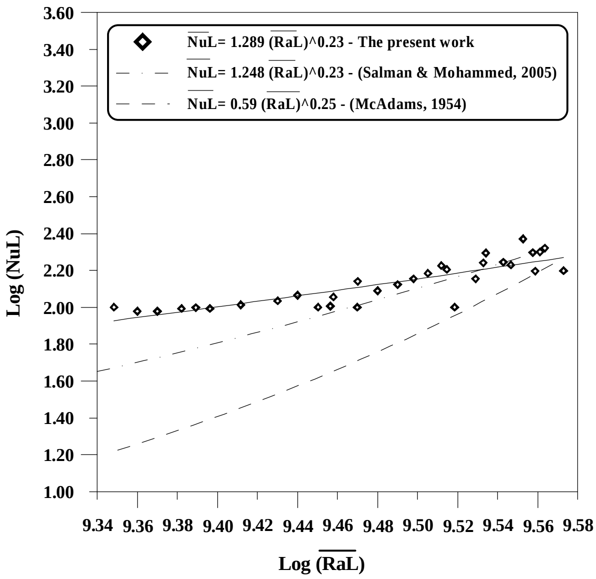

Fig. 17 Correlation of the average heat transfer results and compared with available literature

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انتقال الحرارة بالحمل الحر باستخدام مقاطع مختلفة الاطوال وموضوعةعند مخرج انبوب دائري شاقولي معرض لفيض حراري ثابت

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الخلاصة:

يتناول البحث دراسة عملية لانتقال الحرارة بالحمل الحر لجريان الهواء الطباقي في داخل أنبوب دائري شاقولي باستخدام شرط ثبوت الفيض الحراري ولمدى تغير لرقم رايلي (Ra_L) من (10^8 *6.9) إلى (10^9 *5).ُصمم الجهاز العملي لإيجاد تأثير طول المقيد (restriction) الموضوع في مدخل الأنبوب وفي الموقع العلوي لأنبوب التسخين, على درجة الحرارة على طول سطح الأنبوب المسخن وكذلك على معامل انتقال الحرارة وبالتالي على تغير رقم نسلت الموقعي (Nu_x) والمعدل ($\overline{Nu}$) . الجهاز العملي المستخدم يتكون من أنبوب من الالمنيوم مسخن حرارياً بطول (900mm) وبقطر داخلي (30mm). مقيدات الدخول تتضمن أنبوب دائري أسطواني له نفس القطر الداخلي لأنبوب التسخين ولكن بأطوال(1500 , ١٨٠٠mm, 1200 mm, 900 mm, ,600 mm),

لقد وجد من خلال النتائج العملية أن درجة الحرارة على طول سطح الأنبوب تكون أعلى ما يمكن للمقيد الذي طوله (1800 mm) وتكون اقل ما يمكن للمقيد الذي طوله (1200 mm). أظهرت النتائج أن قيم رقم نسلت الموقعي (Nu_x) والمعدل ($\overline{Nu}$) تكون أعلى ما يمكن للمقيد الذي طوله (1200 mm) وتكون اقل ما يمكن في حالة المقيد الذي طوله (1800 mm). لقد تم الحصول على معادلات تجريبية (empirical equations) بشكل (Log $\overline{Nu_L}$) ضد (Log $\overline{Ra_L}$) لكل حالة من الحالات المستخدمة في البحث وكذلك تم الحصول على معادلة عامة تربط جميع الحالات.