# COMBINED CONVECTION HEAT TRANSFER TO THERMALLY DEVELOPING FLOW IN HORIZONTAL CIRCULAR CYLINDER

**Dr. Yasin K. Salman** **Hussain H. Ahmad** **Hussein A. Mohammed**
Mech. Eng. Dept., College of Engineering, University of Baghdad, Iraq.

## ABSTRACT

Experiments were conducted to study the local and average heat transfer by mixed convection for hydrodynamically fully developed but thermally developing laminar air flow in horizontal circular cylinder. The experimental setup using an aluminum cylinder as test section with 30 mm inside diameter and 900 mm heated length (L/D=30), is subjected to a constant wall heat flux boundary condition. The investigation covers Reynolds number range from (400 to 1600), heat flux varied from (60W/m$^2$ to 400W/m$^2$)and by using an aluminum entrance section pipes (calming sections) having the same inside diameter as test section pipe but with variable lengths as entrance sections. The entrance sections included a long calming section of length 240 cm (L/D=80) and two short calming sections with lengths 60 cm (L/D=20), 120 cm (L/D=40).

The results present the temperature variation along the cylinder surface and the local and average Nusselt number variation with the dimensionless axial distance ($Z^+$). For all entrance sections, the results show an increase in the Nusselt number values as the heat flux increases. Also, the mixed convection regime can be bounded by a suitable selection of (Re) number ranges and the heat flux ranges. The obtained Richardson numbers (Ri) range varied approximately from (0.1171 to 12.54).

## الخلاصة

أجريت تجارب عملية لدراسة انتقال الحرارة الموقعي والمعدل بطريقة الحمل المختلط لجريان الهواء الطباقي تام التطور والمتشكل حرارياً في داخل اسطوانة دائرية أفقية. يتكون الجهاز المستخدم من اسطوانة من الألمنيوم وهي جزء الاختبار بقطر (30mm) وبطول (900mm) نسبة الطول إلى القطر تعادل (L/D= 30) تحت فيض حراري ثابت. يتراوح رقم رينولدز في هذا البحث من (400) إلى (1600) ، أما الفيض الحراري فيتغير من (60W/m$^2$) إلى (400W/m$^2$) ، وباستعمال أنابيب في المدخل ( أنابيب تهدئة ) من الألمنيوم لها نفس القطر الداخلي لجزء الاختبار ولكن بأطوال مختلفة . أنابيب المدخل تتضمن أنبوب مدخل طويل بطول (240cm) (L/D=80) وكذلك أنابيب قصيرة في المدخل وبأطوال (60cm) (L/D=20) و (120 cm)(L/D=40) على التوالي. وضحت النتائج تغير درجة الحرارة على طول سطح الاسطوانة وكذلك تغير رقم نسلت الموقعي ($Nu_x$) والمعدل ($\overline{Nu}$)مع المسافة المحورية غير البعدية ($Z^+$) ، لكل الأطوال المستخدمة ،وبينت النتائج زيادة رقم نسلت كلما زاد الفيض الحراري وكذلك تم الحصول على منطقة الحمل المختلط من خلال الاختيار المناسب لرقم رينولدز و قيم الفيض الحراري. يتغير رقم (Ri) الذي تم الحصول عليه تقريباً من (0.1171) إلى (12.54).

**KEY WORDS**
Combined Convection Heat Transfer, Thermally Developing, Horizontal Circular Cylinder.

## INTRODUCTION

In convective heat transfer problems, the flow usually is classified as forced convection flow in which the flow is caused by external forces such as pumps or fans, or free convection flow in which the flow is created by the fluid density variations due to the wall to fluid temperature difference under the influence of body forces. In most physical applications, the buoyancy forces have negligible effects because they have usually a considerably smaller magnitude than those accompanying the forced flow. But in certain practical situations, however, the magnitude of the two forces may be of the same order and both may then be expected to influence the flow significantly. Therefore, when the free convection superimposed on the forced convection heat transfer process gives rise to new field of study called mixed (combined) convection. Thus, the combined convection situation extends from the extremes of free convection regime on the one hand when the motion results from buoyancy alone, to the forced convection regime on the other hand when external forces alone produce the motion and buoyancy forces are negligible (Metais and Eckert, 1964). Therefore, depending upon the relative magnitude of these two forces, the flow can be divided as pure forced, combined (forced and free) or pure free convection. The interaction of the natural and forced convection currents can be very complex and difficult because it depends not only on all the parameters determining both forced and free convection relative to one another but sometimes also on a large number of interacting parameters including the relative direction of the natural and forced convection to each other, the geometry of the arrangement, the velocity profile at tube entrance and the heating surface boundary conditions.

Laminar flow combined convection heat transfer in tubes is encountered in a wide variety because of special importance in many industrial engineering applications. The following examples can be cited: heating or cooling of heat exchangers for viscous liquids, heat exchangers for gas flows, cooling of electronic equipment, compact heat exchangers, solar collector heat exchangers, the cooling core of nuclear reactors, supercritical boilers and the cooling of rotating parts such as rotor blades of gas turbine also the pipe lines used for transporting oil (Yousef and Tarasuk, 1982). The full understanding of the prevailing velocity and temperature fields, as well as, the pressure drop and heat transfer coefficient, are necessary for the proper design. In addition, to estimate the magnitude of the thermal shock that any one of the preceding systems wall will suffer (Morcos and Bergles, 1975).

The entrance shapes, used in this experimental work, included three entrance sections (calming sections) with different length in which the flow is fully developed at entrance of the heat transfer pipe. For experimental viewpoint, very little investigations have dealt experimentally to study the effect of laminar combined convection to thermally developing flow in a circular cylinder on the heat transfer process.

McComas and Eckert (1966)studied experimentally the fully developed air flow in uniformly heated tube for different ranges of (Re) and (Gr) numbers. The experimental results have revealed that the effect of secondary flow is to reduce the wall to air bulk temperature difference compared with the pure forced convection results in the region far from the tube inlet. Variation of the wall and air temperatures and the local Nusselt number along the tube were presented.

(Mori et al, 1966) carried out experiments to study the effect of buoyancy force on forced convection for fully developed air flow under constant wall heat flux. The velocity and temperature profiles were measured for large (Re.Ra). The calculated Nusselt number was also shown to be twice as those calculated by neglecting the effect of secondary flow. The following correlation formula was obtained:

$$Nu = 0.61(\mathrm{Re}.Ra)^{1/5}\left\{1+\frac{1.8}{(\mathrm{Re}.Ra)^{1/5}}\right\} \qquad (1)$$

Shannon and Depew (1968) performed an experiment to study the influence of free convection on forced laminar flow of water, initially at the ice point in a circular tube with constant wall heat flux and fully developed velocity profile at the onset of heating. The experiment was carried out with different values of (Re), (Gr) and (Gz) numbers. Experiments revealed that the Nusselt number was affected significantly far down stream but relatively little in the thermal entrance region for the tube having X/D=700. A graphical correlation with the parameter $(Gr.Pr)^{1/4}/Nu_{Gz}$ (where $Nu_{Gz}$ is the Nusselt number for pure forced convection) has been achieved giving good agreement with available experimental data.

Depew and August (1971) studied experimentally the fully developed laminar flow in an isothermal tube having an L/D ratio of (28.4). Three different liquids were used in this study: water, ethylalcohol and mixture of glycerol and water. The experiment covered a wide range of (Re), (Gr) and (Gz) numbers. The experimental results show that when dealing with flows in horizontal tubes, the term (Gr.Pr.D/L) does not correctly represents the influence of natural convection for tubes with (L/D) ratios less than 50.

Bergles and Simonds (1971) performed experiments to examine the effects of free convection on laminar water flow in an electrically heated tube having essentially constant wall heat flux. The results have revealed that the natural convection effect may be important even at relatively low Rayleigh number and show that the Nusselt number for developed flow was three times the constant property. Also (Lichtarowicz, 1971) presented the Nusselt number with the product of (Re.Ra) and the temperature profile along the tube was depicted.

(Hong et al, 1974) concluded that for $Ra=10^6$, the Nusselt number in the developing region was more than 300% above the constant property value. The data were correlated accurately by equation, which includes dimensionless groups to account for effects of variable transport properties and tube wall conduction:

$$Nu_{ff} = 0.378\,Gr_{ff}^{\,0.28}\,Pr_{ff}^{\,0.33} \Big/ f^{0.12} \qquad (2)$$

Where: ($f = h.D/k_w * D/t$, t= tube thickness, $k_w$= tube thermal conductivity).

Morcos and Bergles (1975) conducted experiments to investigate the effect of property variation in heated glass and stainless steel tubes with distilled water and ethyleneglycol as test fluids. The measured heat transfer data were presented in a form of correlation:

$$Nu_{ff} = \left[(4.36)^2 + \left\{0.055\left(\frac{Gr_{ff}\,\mathrm{Pr}_{ff}^{1.35}}{P_w^{0.25}}\right)^{0.4}\right\}^2\right]^{1/2} \qquad (3)$$

Yousef and Tarasuk (1982) obtained the average Nusselt number based on the log-mean-temperature difference. The heat transfer results were correlated according to the influence of free convection, which was found to have a significant effect at points close to the tube entrance as follows:

$$Nu\left(\frac{\mu_w}{\mu_b}\right)^{0.14} = 1.75\left[Gz + 0.245\left(Gz^{1.5}.Gr^{1/3}\right)^{0.882}\right]^{1/3} \qquad (4)$$

The purpose of the present investigation is to determine experimentally the effect of Reynolds number and the effect of the heat flux on the laminar air flow heat transfer process under mixed convection situation in uniformly heated horizontal circular cylinder.

## EXPERIMENTAL APPARATUS

The apparatus was constructed to have a test section preceded with different entrance sections, as well as, different Reynolds and Grashof numbers. The open-air circuit, in this investigation, is described first, followed by details of test section and heating element. Then the measuring devices and test procedure is described. Finally, the experimental data analysis method has been presented.

The experimental apparatus shown diagrammatically in **Fig. (1)** is designed and constructed to investigate combined convection heat transfer in a circular cylinder. The apparatus consists essentially of a cylindrical test section as a part of an open-air circuit, mounted on a wooden board (A), which could be rotated around a horizontal spindle.

An open-air circuit was used including small centrifugal fan (F), rotameter(R), test section 'heat transfer pipe' (T) provided with changeable entrance section 'different calming section length'(C). The centrifugal fan derived electrically via-fine control variable resistance so that its power can be regulated accurately. An air control valve (D) was fitted at the fan inlet to obtain fine control of the airflow rate. The air was drawn by the fan in to the test section through the entrance section and then enters the rotameter through flexible hose (M) and then the air leaves the rotameter to the centrifugal fan through another flexible hose (N). Then the heated air was exhausted to the atmosphere.

The test section 'heat transfer pipe' (T) is made of aluminum cylinder with (30) mm inside diameter, (35) mm outside diameter and (900) mm length (L/D=30). The Teflon connection piece (G) represents a part of the test section inlet, with (30) mm inside diameter, (50) mm outside diameter and (80) mm long. Another Teflon piece (I) represents the test section exit and it has dimensions of (30) mm inside diameter, (88) mm outside diameter and (25) mm long. The Teflon was chosen because its low thermal conductivity in order to reduce the test section ends losses.

The air passes through the test section, is fully developed hydrodynamically by using aluminum pipes having same diameter as test section pipe but with variable length as entrance sections. These pipes are connected with the test section by a Teflon connection piece (G) bored with the same inside diameter of the test section and entrance section as shown in

The cylinder is heated electrically by using an electrical heater as shown in **Fig. (1)**, section (A-A). It consists of a (0.5) mm in diameter nickel-chrome wire (H) electrically isolated by ceramic beads, wounded uniformly along the cylinder as a coil with (20) mm pitch in order to give uniform heat flux. An asbestos rope was used as a (20) mm spacer to secure the winding pitch .The outside of the test section was then thermally insulated by asbestos (U) and fiber glass (W) layers, having thicknesses of (15) mm and (13) mm respectively.

The cylinder surface temperatures were measured by twenty-five (0.2) mm-asbestos sheath alumel-chromel (type K) thermocouples, fixed along the cylinder. The measuring junctions (which were made by fusing the ends of the wires together by means of an electric spark in an atmosphere free from oxygen) embed in grooves in the wall normal to the cylinder axis as shown in **Fig. (1)**, section (A-A).

The thermocouples were fixed by drilling twenty-five holes (V) of (1.6) mm diameter and approximately (2) mm deep and along the cylinder wall while the ends of the holes chamfered by a (2) mm drill. The measuring junctions were secured permanently in the holes by sufficient amount of high temperature application Defcon adhesive (X). All thermocouple wires and heater terminals were taken out the test section. Thermocouple positions along the cylinder are shown in **Fig.(1)** section (B-B).

The inlet bulk air temperature was measured by one thermocouple (J) placed in the beginning of the entrance section (calming section), while the outlet bulk air temperature was measured by two thermocouples (K) located in the test section exit 'mixing chamber' (B) .The local bulk air temperature was calculated by fitting straight line -interpolation between the measured inlet and outlet bulk air temperatures.

All thermocouples were used with leads and calibrated using the melting points of ice made from distilled water as reference point and the boiling points of several pure chemical substances.
To perform heat loss calculation through the test section lagging, six thermocouples are inserted in the lagging as two thermocouples at three stations along the heated section (35) cm apart as shown in **Fig. (1)** section (A-A). By using the average measured temperatures and thermal conductivity of the lagging, the heat loss through lagging can be determined.
To evaluate the heat losses from the ends of the test section, two thermocouples were fixed in each Teflon piece. By knowing the distance between these thermocouples and the thermal conductivity of the Teflon, the end losses could be calculated.
Voltage regulator (variac), accurate ammeter and digital voltmeter were used to control and measure of the input power to the working cylinder.
The following entrance sections (calming sections) were used in the present work, that in which the flow is already fully developed at the entrance to the test section. This condition is represented by the pipe with long and short calming sections at entrance as follows: -

1- A long calming section having the same diameter as the test section pipe and length equal to (240) cm (L/D=80) to provide fully developed flow at the entrance of the test section pipe.
2- Two short calming sections, also having the same diameter as the test section pipe and lengths equal to (60) cm (L/D=20) and (120) cm (L/D=40) respectively to provide fully developed flow at the entrance of the test section pipe.

## Experimental Procedure

The procedure employed to carryout a certain experiment was as follows:

1- The required calming section length was fitted with the test section.
2- The centrifugal fan was then switched on to draw the air through the test section while the fan control valve was used for adjusting the required volume flow rate inside cylinder.
3- The electrical heater was switched on and the heater-input power then adjusted to give the required heat flux.
4- The apparatus was allowed to turn on for at least (4 hours) before the steady state conditions were achieved. The readings of all thermocouples were recorded every half an hour by a digital electronic thermometer until the reading became constant, then the final reading was recorded.

The input power to the heater could be changed to cover another run in shorter period of time and to obtain steady state conditions for next heat flux and for same Reynolds number.
The subsequent runs for other Reynolds numbers ranges were conducted in the same procedure.

5- During each test run, the following readings were recorded: -
a- The length of entrance section (calming section) in (cm).
b- The reading of the rotameter (air flow rate) in ($m^3$/hr).
c- The heater current in ampere.
d- The heater voltage in volts.
e- The readings of all thermocouples in ($^{\circ}$C).

## Data Analysis Method

The following simplified steps were used to analyze the heat transfer process for the air flow in a circular cylinder when its surface was subjected to a constant wall heat flux boundary condition.
The total input power supplied to cylinder can be calculated:

$$Q_t = V * I \tag{5}$$

The convection heat transferred from the cylinder surface:

$$Q_{conv.} = Q_t - Q_{cond.} \tag{6}$$

Where $Q_{cond.}$ = is the total conduction heat losses (lagging and ends losses).

The convection heat flux can be represented by:

$$q_{conv.} = \frac{Q_{conv.}}{A_s} \tag{7}$$

Where $A_s = \pi * D * L$

The convection heat flux, which is used to calculate the local and average heat transfer coefficien[t] as follows:

$$h_x = \frac{q_{conv.}}{t_{sx} - t_{ax}} \tag{8}$$

Where: $t_{sx}$= local surface temperature.

$t_{ax}$= local bulk air temperature.

All the air properties were evaluated at the mean film temperature (Louis Burmeister, 1993).

$$t_{fx} = \frac{t_{sx} + t_{ax}}{2} \tag{9}$$

Where: $t_{fx}$= local mean film air temperature.

The local Nusselt number ($Nu_x$) can be determined as:

$$Nu_x = \frac{h_x . D}{k_x} \tag{10}$$

The average values of Nusselt number $(\overline{Nu})$ can be calculated based on the calculated aver[age] surface temperature and average bulk air temperature as follows:

$$\overline{t_s} = \frac{1}{L}\int_{x=0}^{x=L} t_{sx}\, dx \tag{11}$$

$$\overline{t_a} = \frac{1}{L}\int_{x=0}^{x=L} t_{bx}\, dx \tag{12}$$

$$\overline{t_f} = \frac{\overline{t_s} + \overline{t_a}}{2} \tag{13}$$

The average values of the other parameters can be calculated as follows:

$$\overline{Nu} = \frac{qD}{k(\overline{t_s} - \overline{t_a})} \tag{14}$$

$$\overline{Gr} = \frac{g\beta D^3(\overline{t_s} - \overline{t_a})}{\nu^2} \tag{15}$$

$$\overline{Ra} = \overline{Gr} * Pr \quad \text{----------(16)}$$

Where: $\beta = \frac{1}{(273 + \overline{t_f})}$, All the air physical properties $(\rho, \mu, \nu \text{ and } \kappa)$ were evaluated at [the] average mean film temperature ($\overline{t_f}$).

## RESULTS AND DISCUSSION

A total of (48) test runs were conducted to cover the three entrance section pipes with diffe[rent] lengths 60cm (L/D=20), 120cm(L/D=40) and 240cm(L/D=80) for a horizontal circular cylin[der]. The range of heat flux from 60 w/m$^2$ to 400 w/m$^2$ and Reynolds number varied from (400 to 160[0]

**Surface Temperature Distribution**

Generally, the variation of the surface temperature along the cylinder may be affected by many variables such as heat flux, Reynolds number and the flow entrance situation. The temperature variation for selected runs is plotted in **Figs. (3 - 6)**.

**Fig. (3)** shows the variation of the surface temperature along the cylinder for different heat flux, for Re=400 and for calming section length equal to 60 cm (L/D=20). This figure reveals that the surface temperature increases at cylinder entrance to reach a maximum value after which the surface temperature decreases. The location of maximum temperature seems to move toward the cylinder entrance as the heat flux increases. This can be attributed to the developing of the thermal boundary layer faster due to buoyancy effect as the heat flux increases for the same (Re).

**Fig. (4)** is similar to **Fig. (3)** but pertains to Re=1600.The curves in the two figures show same trend, but the surface temperature values in **Fig. (4)** are lower than values in **Fig. (3)** because of the forced convection domination.

**Figs. (5&6)** show the effect of (Re) variation on the cylinder surface temperature for low heat flux (92 w/m$^2$) in **Fig. (5)** and for high heat flux (294 w/m$^2$) in **Fig. (6)**. It is obvious that the increasing of (Re) reduces the surface temperature, as the heat flux kept constant. It is necessary to mention that as heat flux increases the surface temperature increases because the free convection is the dominating factor in the heat transfer process.

The surface temperature variation for the second calming section with length equal to 120 cm (L/D=40) and third calming section with length equal to 240 cm (L/D=80), is similar trend as mentioned for (L/D=20).

**Local Nusselt Number Distribution ($Nu_x$)**

The variation of the local Nusselt number ($Nu_x$) with the dimensionless axial distance ($Z^+$), is plotted for selected runs in **Figs. (7 - 10)**.

**Figs. (7&8)** show the effect of the heat flux variation on the ($Nu_x$) distribution for Re=400 and Re=1600 respectively. It is clear from these two figures that at the higher heat flux, the results of ($Nu_x$) were slightly higher than the results of lower heat flux. This may be attributed to the secondary flow superimposed on the forced flow effect increases as the heat flux increases leading to higher heat transfer coefficient. The dotted curve in each figure represents the theoretical pure forced convection (TPFC) based on constant property analysis of (Shah and London, 1978).

**Figs. (9&10)** show the effect of (Re) number variation on the $Nu_x$ distribution with ($Z^+$), for low heat flux (92 w/m$^2$) in **Fig. (9)** and for high heat flux (294 w/m$^2$) in **Fig. (10)**. For constant heat flux, the (Nu) values give higher results than the predicted pure forced convection value and moves toward the left as (Re) increases. This situation reveals that the forced convection is dominant on the heat transfer process with little effect of buoyancy force for high (Re). As the (Re) number reduced, the buoyancy effect expected to be higher which improves the heat transfer results.

It is necessary to mention that in horizontal cylinder, the effect of secondary flow is high, hence at low (Re) number and high heat flux, situation makes the free convection predominant. Therefore, as the heat flux increases, the fluid near the wall becomes warmer 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 cylinder center flows downstream. This sets up two longitudinal vortices, which are symmetrical about a vertical plane. These vortices reduces the temperature difference between the cylinder surface and the air flow in which led to increase the growth of the thermal boundary layer along the cylinder and causes an improvement in the heat transfer results. But at low heat flux and high (Re) number the situation makes the forced convection predominant and the vortex strength decreases which decreases the temperature difference between the surface and the air, hence the ($Nu_x$) values becomes higher (Mori and Futagami, 1967).

Also, the ($Nu_x$) distribution for higher calming section (L/D=40,L/D=80) are similar trend as mentioned in the first calming section length (L/D=20).

**Average Nusselt Number Distribution** $(\overline{Nu})$

The variation of the average Nusselt number $(\overline{Nu})$ with the dimensionless axial distance $(Z^+)$ is depicted for selected runs in **Figs. (11 - 14).**

**Figs. (11&12)** show the effect of the heat flux on the $(\overline{Nu})$ for Re=400 and Re=1600 respectivel and the effect of (Re) number on the $(\overline{Nu})$ for low heat flux (92 $w/m^2$) and high heat flux (29 $w/m^2$) in **Figs. (13&14)** respectively for the shorter tube (L/D=20). The $(\overline{Nu})$ variation for highe calming section (L/D=40,L/D=80) are similar trend as mentioned for (L/D=20).

## CONCLUSIONS

As a result from the experimental work conducted in the present investigation to study combine convection heat transfer to thermally developing laminar air flow in horizontal circular cylinde subjected to a constant wall heat flux boundary condition, the following conclusions can be drawn:

1- The variation of the surface temperature along the cylinder has the same shape. This variation i affected by:

a- The surface temperature increases as the heat flux increases, for the same (Re).

b- The surface temperature for low (Re) is higher than for high (Re), for the same heat flu because of the free convection domination.

2- The variation of $(Nu_x)$ with $(Z^+)$, for all entrance lengths has the same trend. This variation i summarized as follows:

a- For the same (Re) number, the $(Nu_x)$ increases with the increase of heat flux.

b- For the same heat flux and high (Re), the $(Nu_x)$ moves toward the left of the $(Nu_x)$ predicted f (TPFC), because the forced convection is dominant.

c- For the same heat flux and low (Re), the $(Nu_x)$ moves toward the right of the $(Nu_x)$ predicte for (TPFC), because the natural convection is dominant.

3- Free convection effects tended to decrease the heat transfer results at low (Re) and to increas the heat transfer results for high (Re).

4- At the cylinder entrance the effect of buoyancy is small, but it is effects increase in the cylind downstream.

5- The variation of the $(\overline{Nu})$ with $(Z^+)$, has the same heat transfer characteristics which mentioned i the $(Nu_x)$ results.

6- The mixed convection regime can be bounded by a suitable selection of (Re) number ranges an the heat flux ranges. The obtained Richardson numbers (Ri) range is varied approximately fro (0.1171 to 12.54).

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**NOMENCLATURE**

| Symbol | Description | Unit |
|---|---|---|
| $A_s$ | Cylinder surface area | $m^2$ |
| Cp | Specific heat at constant pressure | J/kg.°C |
| D | Cylinder diameter | m |
| g | Gravitational acceleration | $m/s^2$ |
| h | Heat transfer coefficient | $w/m^2.°C$ |
| I | Heater current | amp. |
| k | Thermal conductivity | w/m.°C |
| L | Cylinder length | m |
| $Q_{cond}$ | Conduction heat loss | w |
| $q_{conv}$ | Convection heat flux | $w/m^2$ |
| $Q_{conv}$ | Convection heat loss | w |
| $Q_t$ | Total heat input | w |

| | | |
|---|---|---|
| R | Cylinder radius | m |
| t | Air temperature | $^{\circ}C$ |
| $t_1$ | Wall thickness | m |
| V | Heater voltage | volt |
| x | Axial distance | m |

**Greek**

| | | |
|---|---|---|
| $\beta$ | Thermal expansion coefficient | 1/K |
| $\mu$ | Dynamic viscosity | kg/m.s |
| $\nu$ | Kinematic viscosity | $m^2/s$ |
| $\rho$ | Air density | $kg/m^3$ |

**Dimensionless Group**

| | | |
|---|---|---|
| Gr | Grashof number | $= g \beta D^3 (t_s - t_a)/\nu^2$ |
| Gz | Graetz number | = Re.Pr.D/L |
| Nu | Nusselt number | = h.D/ k |
| Pr | Prandtl number | $= \mu \, . \, Cp/k$ |
| $P_w$ | Wall parameter | $= h.D^2/k_w \, . \, t_1$ |
| Ra | Rayleigh number | = Gr.Pr |
| Re | Reynolds number | $= \rho \, . \, v. \, D \, / \, \mu$ |
| Ri | Richardson number | $= Gr/Re^2$ |
| $Z^+$ | Axial distance | = x/D.Re.Pr |

**Subscript**

| | |
|---|---|
| a | Air |
| b | Bulk |
| calm. | Calming section |
| f | Film |
| f | Fully developed flow |
| s | Surface |
| w | Wall |
| x | Local |

**Superscript**

| | |
|---|---|
| —— | Average |

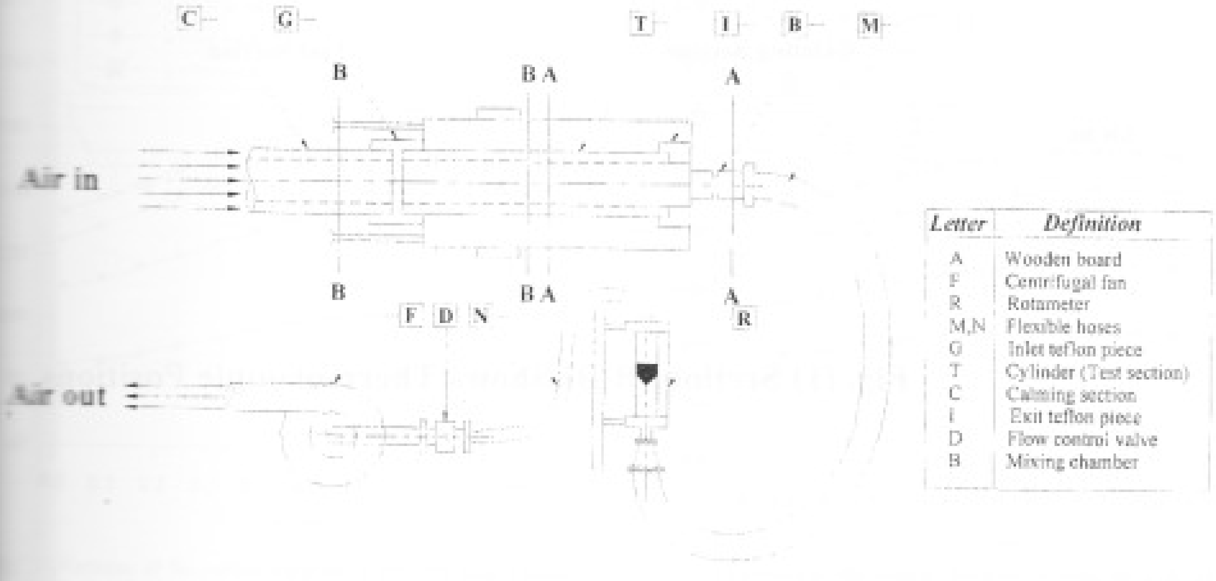
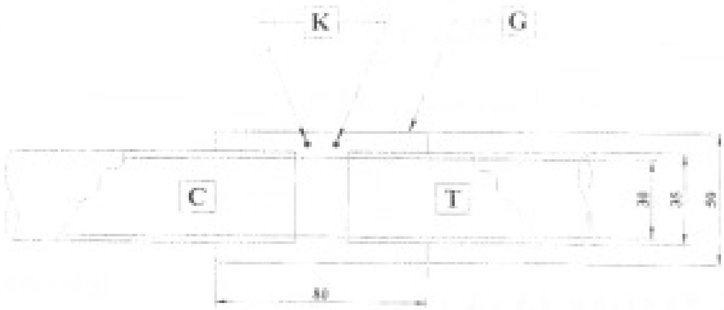



**Fig. (1) Diagram of Experimental Arrangement**


**Fig.(1) Section (A-A)**

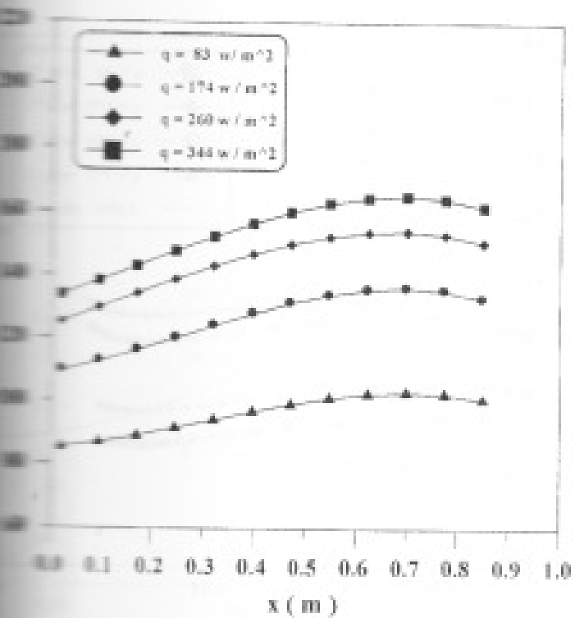

**Fig. (1) Section (B-B) Shows Thermocouple Positions**




| *Letter* | *Definition* |
|---|---|
| T | Test section |
| C | Calming section |
| G | Teflon connection piece |
| K | Thermocouple |

*All Dimensions in (mm)*

**Fig.(2) Teflon Connection Piece.**




Variation of the surface temperature with the axial distance for Re=400 , Lcalm.=60 cm , L / D)calm.=20

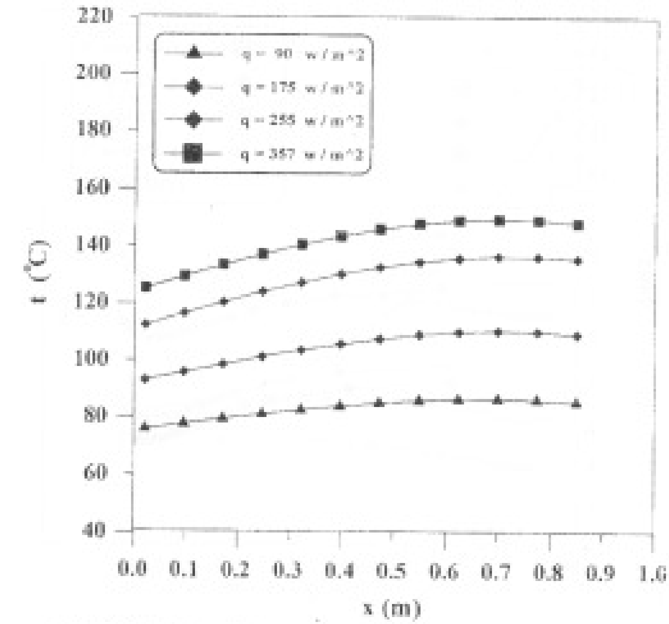
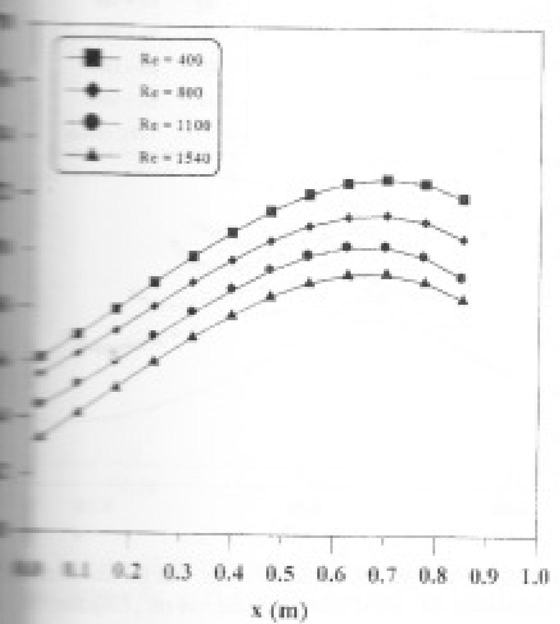



Fig.( 4 ) Variation of the surface temperature with the axial distance for Re=1600 , Lcalm.=60 cm , L/D) calm.=20




Variation of the surface temperature with the axial distance for q=92 w/m^2 , Lcalm.= 60 cm , L/D)calm.= 20

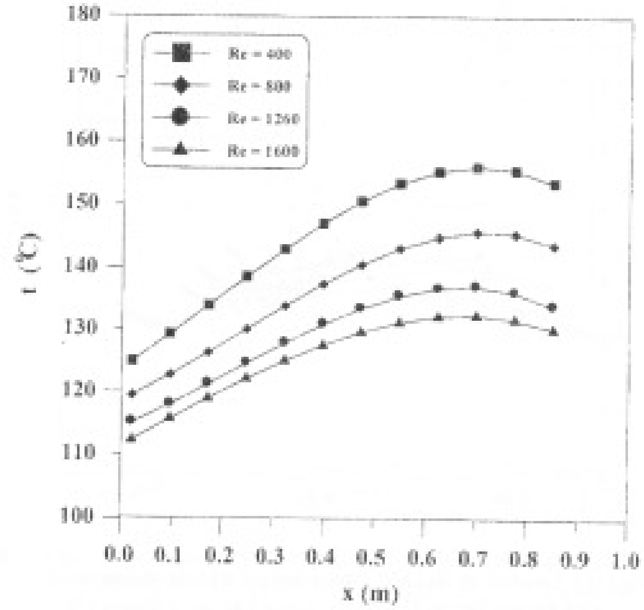



Fig.( 6 ) Variation of the surface temperature with the axial distance for q=296 w/m^2 , Lcalm.= 60 cm , L/ D)calm.= 20

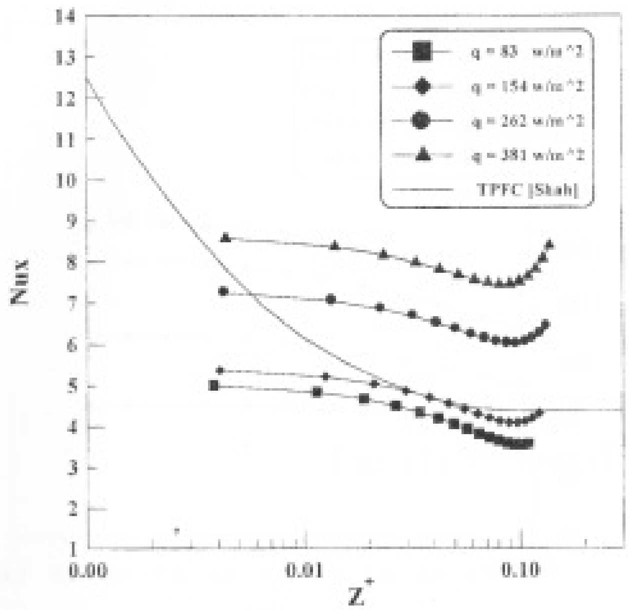



Fig.( 9 ) Variation of the Local Nusselt number with the dimensionless axial distance for Re = 400 , Lcalm.= 60 cm , L/D)calm.=20

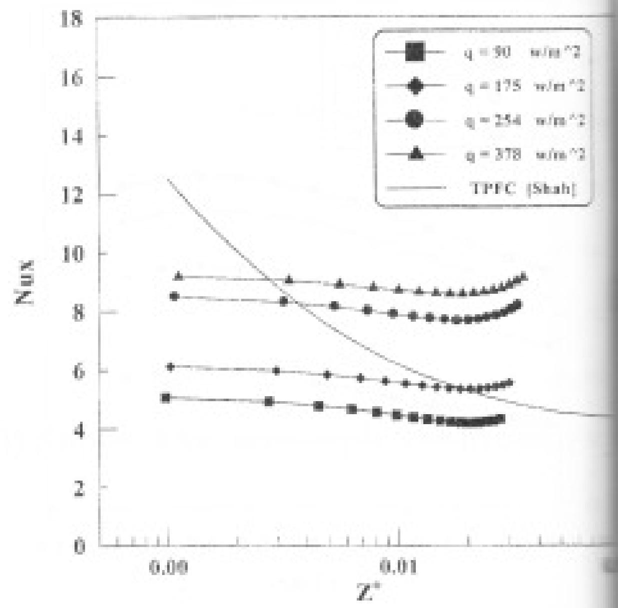



Fig.(10 ) Variation of the Local Nusselt number with the dimensionless axial distance for Re = 1600 , Lcalm.= 60 cm , L/D)calm.=20

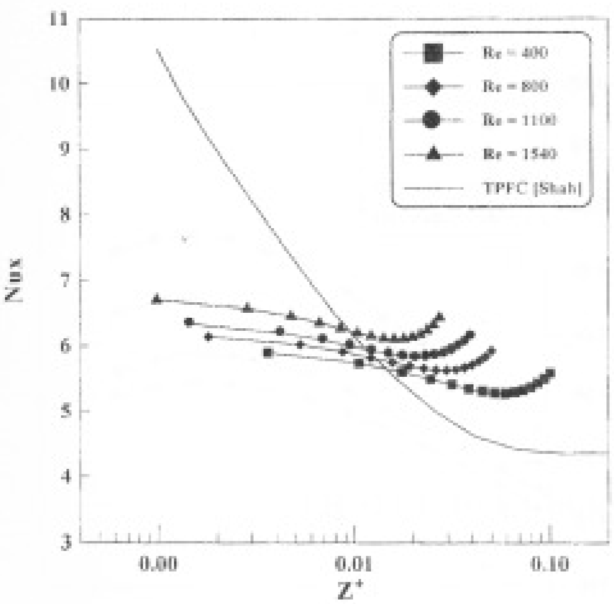



Fig.( 11 ) Variation of the Local Nusselt number with the dimensionless axial distance for q= 92 w/m^2 , Lcalm.= 60 cm , L/D)calm.=20

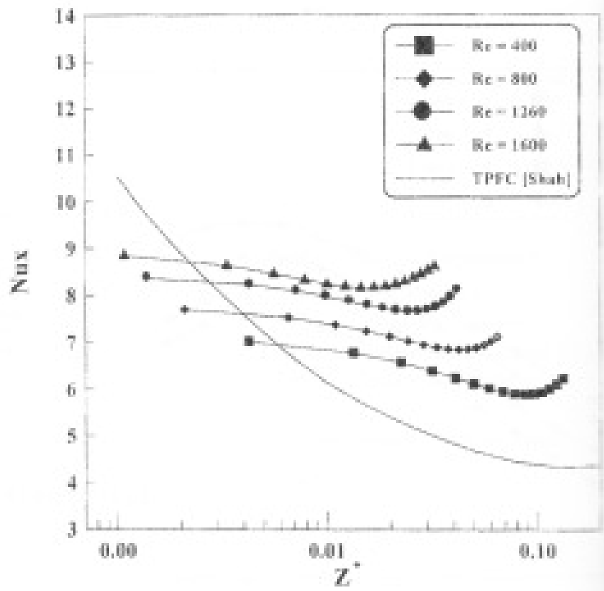
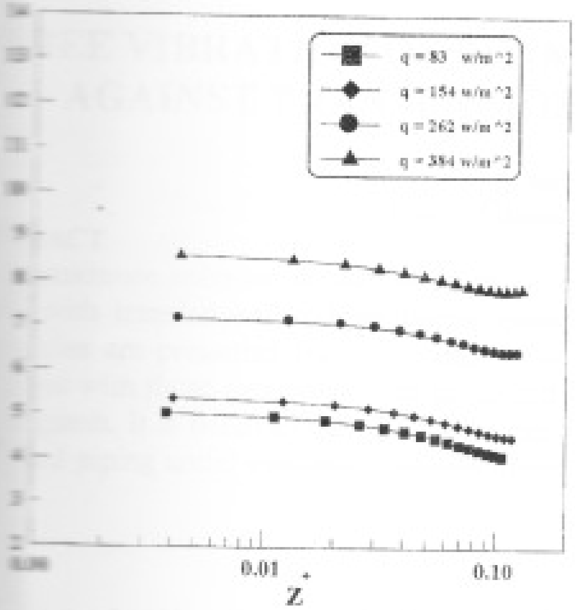



Fig.( 12 ) Variation of the Local Nusselt number with the dimensionless axial distance for q=294 w/m^2 , Lcalm.= 60 cm , L/D)calm.=20




Variation of the average Nusselt number with the dimensionless axial distance for Re= 400, Lcalm.= 60 cm , L/D)calm.=20

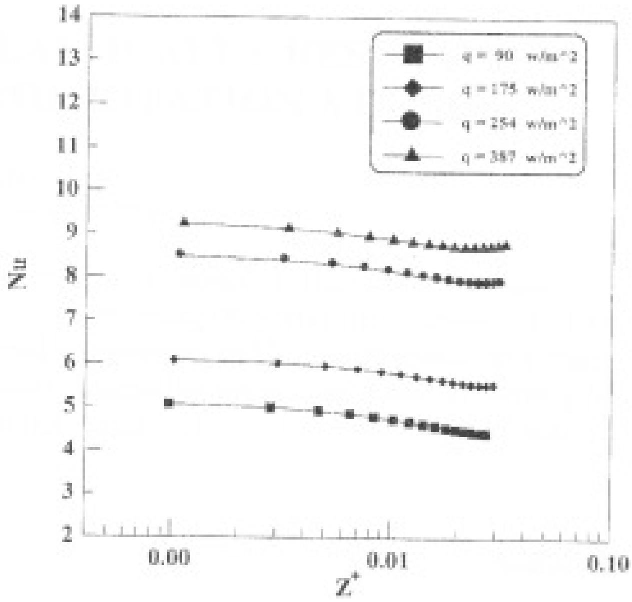
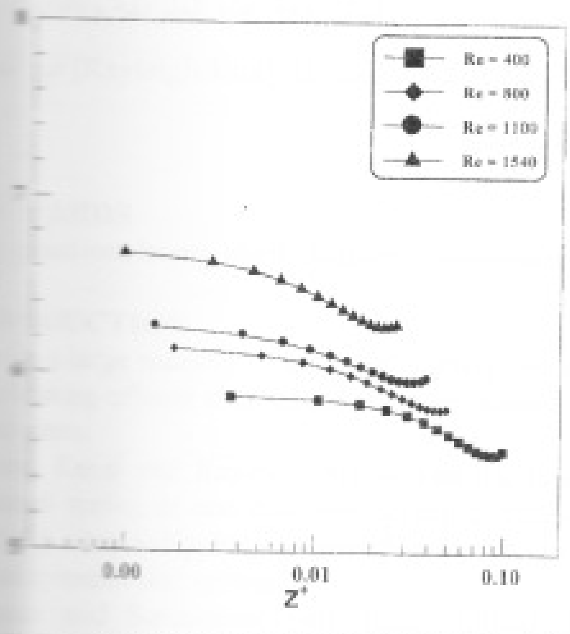



Fig.( 17 ) Variation of the average Nusselt number with the dimensionless axial distance for Re= 1600, Lcalm.= 60 cm , L/D)calm.=20




Variation of the average Nusselt number with the dimensionless axial distance for q= 92 w/m^2, Lcalm.= 60 cm , L/D)calm.=20

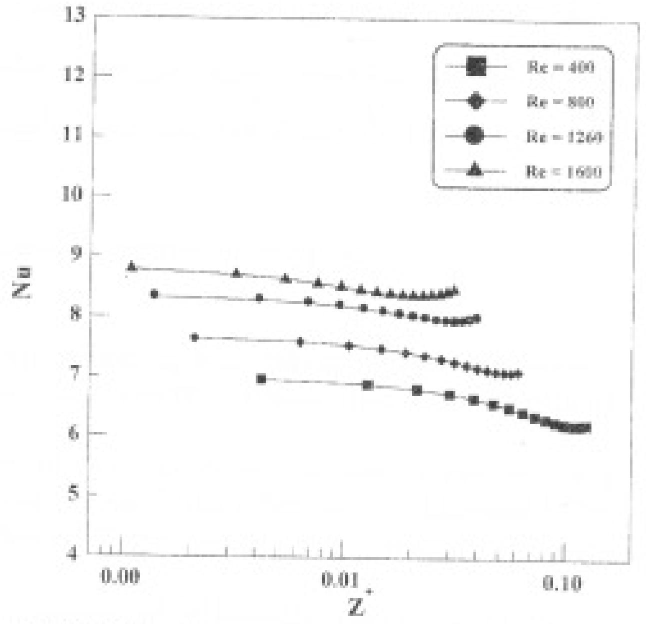



Fig.( 19 ) Variation of the average Nusselt number with the dimensionless axial distance for q= 294 w/m^2, Lcalm.= 60 cm , L/D)calm.=20