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# Combined Convection Heat Transfer at the Entrance Region of Horizontal Semicircular Duct

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*Abstract*— **Experimental study has been conducted on mixed convection heat transfer at the entrance region of uniformly heated horizontal semicircular duct with flat plate at the bottom. The effect of heat flux and Reynolds number on the local and average Nusselt number has been investigated. The Study has been covered a wide range of heat flux (510 $\leq$q $\leq$ 1060) W/m2 and Reynolds number varies from 500 to 2000 . The results show that, the local Nusselt number value increases as Reynolds number and heat flux increase. An empirical equation of average Nusselt number as a function of Rayleigh number and Reynolds number has been deduced.**

*Index Terms*— **combined convection, heat transfer, semicircular duct.**

## I. INTRODUCTION

Due to the prominent importance of heat transfer in energy technology, several practical applications involving mixed convection in ducts of various cross-sections and orientations continue to command substantial attention. Full understanding of the prevailing velocity and temperature fields, as well as the pressure drop and heat transfer characteristics are necessary for the proper design of such systems. The combined convection heat transfer inside semicircular duct is important in energy technology such as solar energy, cooling of electronic components, compact heat exchangers and the cooling core of nuclear reactors. Many anthers studied experimentally and theoretically the mixed convection process inside noncircular duct. **Etemad et al. (1997), [1],** used distilled water and viscous non Newtonian fluids described by the law power model flowing in straight channel of equilateral triangular and semicircular cross-sections. They concluded that the values of local Nusselt number depends strongly on Rayleigh number. **Busedra and Soliman (1999), [2],** used water as a working test to study the combined convection at the entrance region of a uniformly heated semicircular duct with (aiding and opposing flow direction ($\alpha = \pm 20^{\circ}$). Results show that for the angle up to 20°, the effect of Reynolds number was small and this effect increases at the downward inclination angles. **Muzychka et al (2004), [3]**, studied numerically the combined convection at the entrance region of noncircular ducts and channels. This model predicted both local and average Nusselt numbers and was valid for both isothermal and constant heat flux boundary conditions. The model was developed using the asymptotic results for convection from a flat plate, thermally developing flows in non-circular ducts, and fully developed flow in non-circular ducts. Comparisons were made with several existing models for the circular tube and parallel plate channel and with numerical data for several non-circular ducts. **Ben-Arous and Busedra (2008) [4],** studied numerically the combined convection in horizontal semicircular ducts (flat wall at the bottom) with radial internal fins. The wall of the duct was assumed to have a uniform heat input along the axial direction with a uniform peripheral wall temperature. The analysis focused on the case of hydrodynamically and thermally fully-developed laminar flow. The governing equations for the velocity and temperature were solved by using a control volume based finite difference approach. The fluid flow and heat transfer characteristics were found to be dependent on the Grashof number, the fin length and the number of fins. It was also found that the heat transfer rate increases with the Grashof number and was more intense with respect to the case of finless semicircular ducts. The most remarkable outcome of the present study was that, for each number of fins. An optimum fin length exists at which the Nusselt number attains a maximum. **Etemad et al. (2009), [5]** studied a numerical investigation used the finite element method to solve the full three-dimensional governing equations deal with simultaneously developing laminar flow triangular ducts with viscous non-Newtonian test fluid. The boundary condition was taken, constant heat flux and constant wall temperature and it was shown that the Nusselt number distribution along the surface is effected appreciably by the variation of the power law.

**Manar et al (2012),[6],** studied the fully develop forced conviction in a triangular channel at horizontal position with uniform heat Flux as a boundary condition. The ranges of heat Flux and Reynolds number were ($91 \leq q \leq 409$) W/m$^2$ and $1198 \leq Re \leq 1988$ . Results show. That the average transfer coefficient is maximum at the beginning of the channel the decreases downstream because of the growing thermal boundary layer along the channel .The result s was compared with other works and gave good agreement.. **Ahmed et al (2013), [7]** studied the mixed convection heat transfer and the laminar flow in an inclined rectangular cylinder with uniform heat flux from upper surface (40 to 500 W/m$^2$) and insulated other surfaces. The angle of inclination were (-30°, -45° and -60°). The modified Grashof number varied from $2.38\times10^6$ to $2.8\times10^8$ while the Reynolds number varied from 455 to 2000. They concluded the heat transfer process depends strongly on Richardson number. **Busedra and Soliman (2015), [8]**, studied the laminar, fully developed mixed convection in an inclined semicircular ducts under buoyancy-assisted and buoyancy-opposed condition. A numerical finite control volume approach was adopted in solving the governing equations, and results were obtained for the two limiting boundary conditions $h_1$ and $h_2$. These results included the

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velocity and temperature distributions, a map for the onset of flow reversal, and data for the friction factor and Nusselt number. The results were presented with a detailed assessment of the effects of inclination, Reynolds number, Grashof number, and the thermal boundary Conditions **.Rajamohan1 et al (2015), [9]**, studied the convection heat transfer process by mixed combined, at thermally developing region in an inclined heated square duct with one side heated isothermal and three adiabatic wall with the range of hot wall temperature from 30°C to 100°C, heat flux from 252 to 872 W/m$^2$, and Reynolds number range $858 \leq Re \leq 1788$. The surface wall emissivity was considered to be 0.05 and 0.85. Flow visualizations to obtain the flow structure of free convection and mixed convection were carried out. The result showed that the angle of inclination, Reynolds number and radiation surface have important effects on that the heat transfer enhancement and local Nusselt number... **Busedra and Budabous (2016), [10],** studied numerically the development of laminar mixed convection with heat and mass transfer in vertical and horizontal semicircular cylinder for the case of uniform heat input boundary condition, concentration at the fluid–solid interface axially, and uniform peripheral wall temperature at any axial station for Pr=0.7, Re=500, and Gr=$1.66_x10^5$. The results showed that the forced-convection boundary layer development dominates very close to the duct inlet, while further downstream; the heat and mass transfer rates are enhanced due to the effect of solute buoyancy.

The purpose of this work to study the mixed convection heat transfer at the entrance region of semicircular cylinder, and deduce an empirical equation of mean Nusselt number as a function of Reynolds number and Rayleigh number.

## II. Experimental Facilities--

The experimental apparatus shown in Figure (1) schematically consists essentially of two parts: flow measuring section and test section. The test section is a part of open air loop circle. The test section is mounted on a wooden board (K), and the table made from iron which can be rotated around a horizontal spindle (L). The inclination angle of the semicircular cylinder can thus be adjusted as required. The air is induced by a centrifugal fan (A) , enters the orifice pipe section (Standard British Unit) (B) and then settling chamber (C) through a flexible hose (D) then test section. The settling chamber was carefully designed to reduce the flow fluctuation and to get a uniform flow at the test section entrance by using flow straightener (E) and to get uniform flow. The flow passes then into the calming section (F) through a well-designed cone

(G) to prevent the disturbance at the entrance. The inlet air temperature was measured by one thermocouple (J) located in the settling chamber (D) while the outlet bulk air temperature was measured by three thermocouples (Z) located in the test section exit. The local bulk air temperature was calculated by using a straight line interpolation between the measured inlet and outlet bulk air temperature. Semicircular Teflon piece (M) with the same dimensions of test section is fitted at the exit of test section. Teflon material has been chosen because of its low thermal conductivity in order to reduce the heat losses from the aluminum cylinder ends.

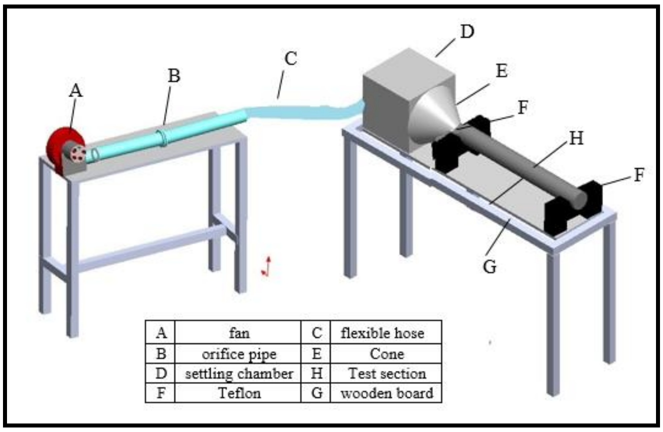
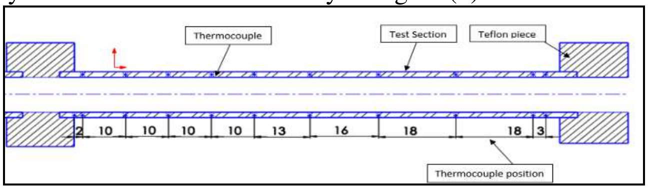
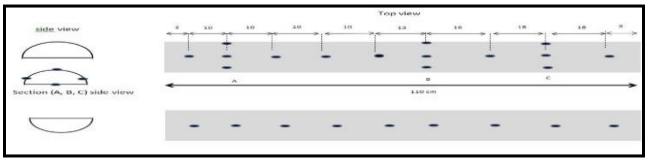



Figure. (1): Schematically of experimental apparatus

Twenty four aluminum –chrome thermocouples (type k) with 2m length have been fixed along and around the semicircular cylinder as shown schematically in Figure (2).




a) Locations of thermocouples along test section




b) Top and side view for test section shows position of thermocouples Figure. (2) :Schematic of thermocouple positions on test section

## III. Heating Element

The heat input in the test section was generated by nickel chrome electric resistance wire with a total resistance of 49 Ω covered by insulating beads and then wrapped by a layer of asbestos rope to reduce the heat losses. Six thermocouples are fixed along the outer surface of duct (three thermocouples for each upper and lower surface. The heater circuit consists of a variac to adjust the heater input power as required while a digital ammeter and voltmeter were used to measure the heater current and voltage. An asbestos was used as insulation for test section as shown schematically in Figure (3).

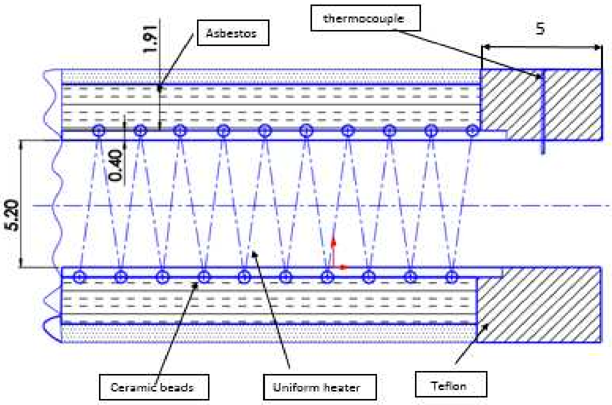



Figure. (3): Schematic of heating element.

To determine the heat loss from the test section ends, three thermocouples were fixed on outer surface insulation Teflon piece. The distance between these thermocouples was 1 cm. Knowing the thermal conductivity of the Teflon; the ends losses could thus be calculated

Simplified steps are used to analyze the heat transfer process for the air flow in semicircular cylinder with a uniform heat flux. The total input power supplied to the semicircular cylinder can be calculated:

$Q_t = V \times I$ ………………………………….(1)

The convection and radiation heat transferred from the inner cylinder is:

$Q_{cr} = Q_t - Q_{cond}$ …………………………..…….. (2)

Where $Q_{cond}$ is the conduction heat loss which was found experimentally equal to 3 % of the input power.The convection and radiation heat flux can be represented by:

$q_{cr} = Q_{cr}/A$ …………………………………..….(3)

Where:

$A = \pi r L + 2rL$

The convection heat flux, which is used to calculate the local heat transfer coefficient is obtained after deduce the radiation heat flux from $q_{cr}$ value.

The local radiation heat flux can be calculated as follows:

$q_r = F_{1-2} \ \sigma \ \epsilon [((t_s)_z + 273)^4 - ((t_b)_z + 273)^4]$.. ……..(4)

Where:

$(t_s)_z$ = local temperature of outer surface of semicircular duct

$\epsilon$ = emissivity of the polished aluminum surface=0.09.

$F_{1-2} \approx 1$

Hence the convection heat flux at any position is:

$q = q_{cr} - q_r$ ....................................... (5)

The local heat transfer coefficient can be obtained as:

$$h_z = \frac{q}{(t_s)_z - (t_b)_z} \quad \text{……………………………….(6)}$$

$(t_b)_z$ = Local bulk air temperature.

All the air properties are evaluated at the mean film air temperature [11].

$$(t_f)_x = \frac{(t_s)_x + (t_b)_x}{2} \quad \text{………………………….(7)}$$

$t_f$ = Local means film air temperature.

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

$$Nu_x = \frac{h_x \quad D_h}{\kappa} \quad \text{………………………… (8)}$$

The average values of Nusselt number $Nu_m$ can be calculated based on calculation of average inner surface temperature and average bulk air temperature as follows:

$$\overline{t_s} = \frac{1}{L} \int_{z=0}^{z=L} (t_s)_x \quad dz \quad \text{................(9)}$$

$$\overline{t_b} = \frac{1}{L} \int_{x=0}^{x=L} (t_b)_x \quad dx \quad \text{……..... (10)}$$

$$\overline{t_f} = \frac{\overline{t_x} + \overline{t_b}}{2} \quad \text{……………….(11)}$$

$$Nu_m = \frac{q \quad D_h}{\kappa(\overline{t_x} - \overline{t_b})} \quad \text{……… (12)}$$

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

$$Re_m = \frac{\rho \quad u_i \quad D_h}{\mu} \quad \text{…….. (13)}$$

$$Gr_m = \frac{g \quad \beta \quad D_h^3 \quad (\overline{t_x} - \overline{t_b})}{\nu^2} \quad \text{…….. (14)}$$

$$Pr_m = \frac{\mu \quad C_p}{k} \quad \text{…………… (15)}$$

$Ra_m = Gr_m \cdot Pr_m$ ………….. (16)

Where:

$$\beta = 1/(273 + \overline{t_f})$$

All the air physical properties $\rho$, $\mu$, $\nu$, and k were evaluated at the average mean film temperature ( $\overline{t_f}$ ) [12].

## IV. RESULTS

The experimental part covered a wide range of Reynolds number (Re varied from 500 to 2000) with heat flux varied from 522 W/m$^2$ to 1060 W/m$^2$. In general, there are many variables which have affect the surface temperature along the cylinder such as velocity of the flow (Reynolds number), the heat flux, the velocity profile at entrance.

### Temperature Variation

Figures (4 and 5) show the effect of Renolds number on the heat transfer with constant heat flux (q=1035 and 537 W/m$^2$ ).It can be seen that the surface temperature decreases as the (Re) value increases because of the significant forced convection to remove heat from the semicircular cylinder surface. These figures show that for a given value of Reynolds number both the local duct surface and air temperatures increase with increasing the axial distance. Figures (6 and 7) show the variation of temperature distribution along the semicircular duct surface for various values of heat flux and Re = 2000 and 500, respectively. Figure. (6) show that the surface temperature at the entrance of the test section begins gradually to increase up until reaches a maximum value at the duct exit. Also, The values of temperature increases as heat flux increases because of the dominant buoyancy effect in the heat transfer process as the heat flux increases for the same Reynolds number. Figure. (7) shows the same behavior obtained in Figure. (6) but with a higher value of temperature because the high Reynolds number rise the forced convection.

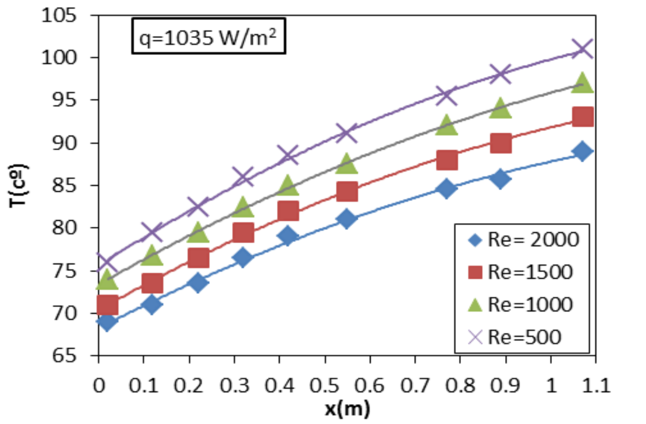



Figure (4): Influence of Reynolds number on distributions of axial surface Temperatures q=1035 W/m$^2$.

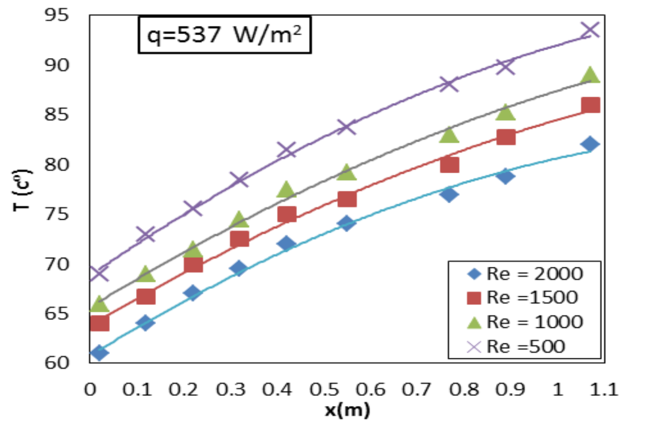



Figure (5): Influence Reynolds number on distributions of axial surface temperatures for q=537 W/m$^2$.

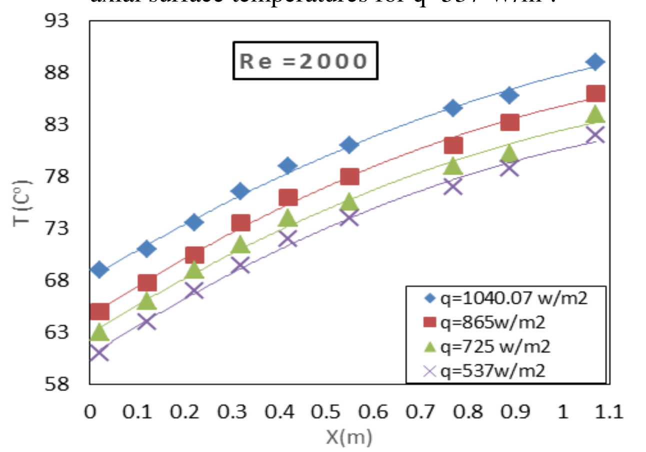



Figure (6):Influence of heat flux on distribution of axial surface temperatures for Re=2000.

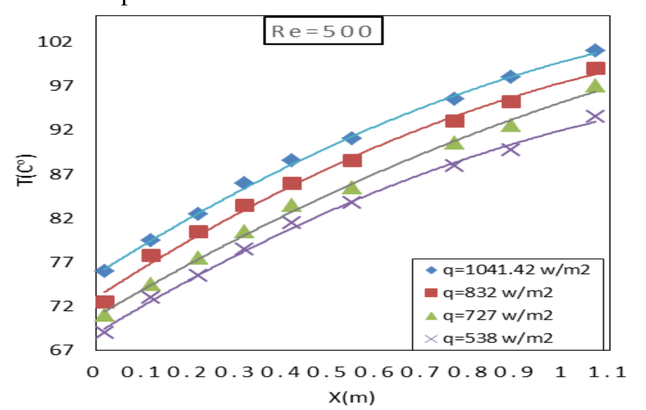



Figure (7): Influence of heat flux on distribution of axial surface temperatures for Re=500.

## Local Nusselt Number ($Nu_x$)

Generally, the local Nusselt number ($Nu_x$) decreases upstream until it reaches a minimum value due to the high temperature difference between the surface and the air at this region. Then Nusselt number increases downstream because the strong secondary currents will be strong at this region leads to improve heat transfer coefficient. Figures (8 and 9) show the effect of Reynolds number on the $Nu_x$ distribution with the logarithmic dimensionless axial distance (logarithmic inverse Greatz number log ZZ) for q = 1034 W/m$^2$ and 537 W/m$^2$; respectively. It is obvious from these figures that the (Nux) distribution is higher for higher (Re) than that for lower (Re) because of the dominant forced convection in the heat transfer process with a little effect of the buoyancy force.

Figures (10 and 11) shows the effect of the heat flux on the Nux distribution for Re = 2000 and Re=500; respectively. It is clear from this figure that the local Nusselt number values increase as heat flux increases. This related to that the increasing of heat flux leads to increase the effect of secondary flow, which enhances the heat transfer coefficient. It is necessary to mention that at the horizontal position, the effect of secondary flow is high, hence at low Reynolds number and high heat flux, the situation makes the free convection predominate. 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 sidewalls, and by continuity, the fluid near the semicircular cylinder core flows downstream. This sets up an expected two longitudinal vortices which are symmetrical about a vertical plane. These vortices reduce the temperature difference between the cylinder surface and the airflow, and this leads to an improvement in the heat transfer results. But, at low heat flux and high (Re), the situation makes the forced convection predominate and the vortices strength diminishes and this allows the forced flow to decrease the temperature difference between the surface and the air, hence the (Nux) values become higher. The curves move to the left as the Reynolds number increases because of the dominating of forced convection with little effect of the buoyancy force.

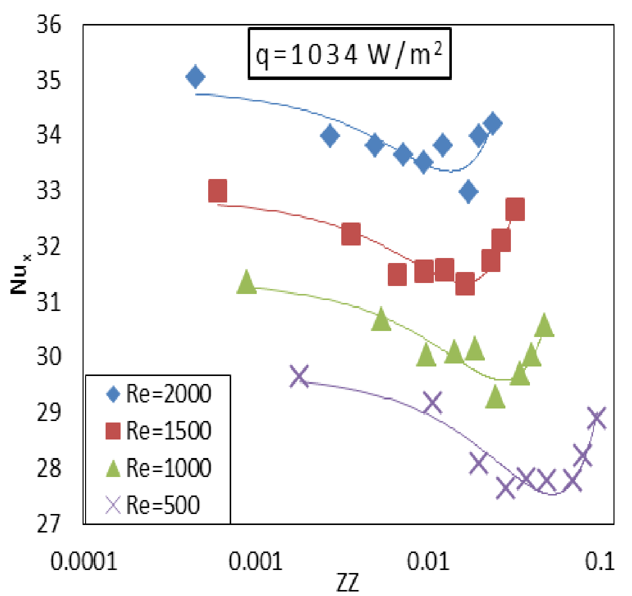



Figure (8): Influence of (Re) on local Nusselt number for q=1034 W/m$^2$.

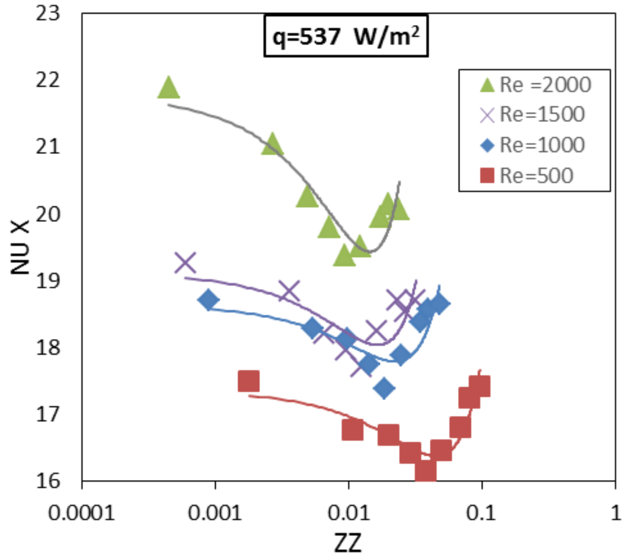



Figure (9) Influence of (Re) on local Nusselt numbers for q=537 W/m$^2$.

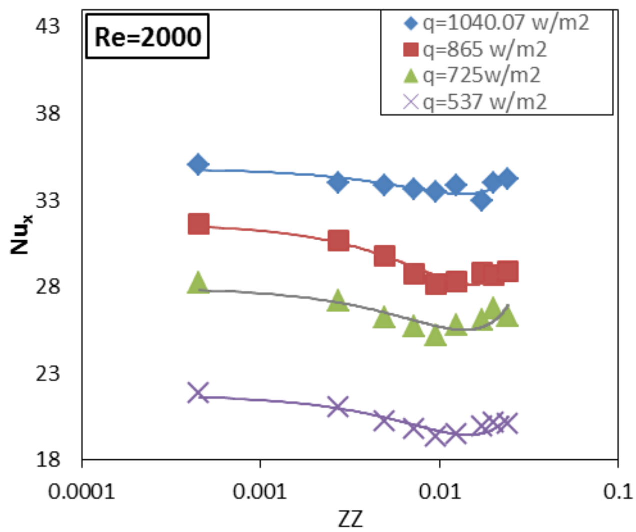



Figure (10): Influence of heat flux on local Nusselt numbers for Re= 2000.

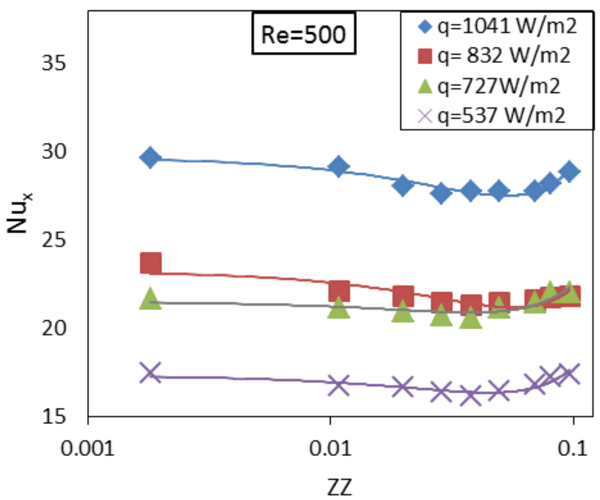



Figure (11) :Influence of heat flux on local Nusselt numbers for Re= 500.

**Average Nusselt Number Versus Reynolds Number**

Figure (12) show the variation of mean Nusselt number versus Reynolds number for various values of heat flux . It can have noticed from this figure that the average Nusselt number increases linearly with increasing of Reynolds number at the same heat flux because of dominant forced convection in the heat transfer process. Also, the average Nusselt number value increases with increasing of heat flux for the same Reynolds number because of dominant free convection in the heat transfer process.

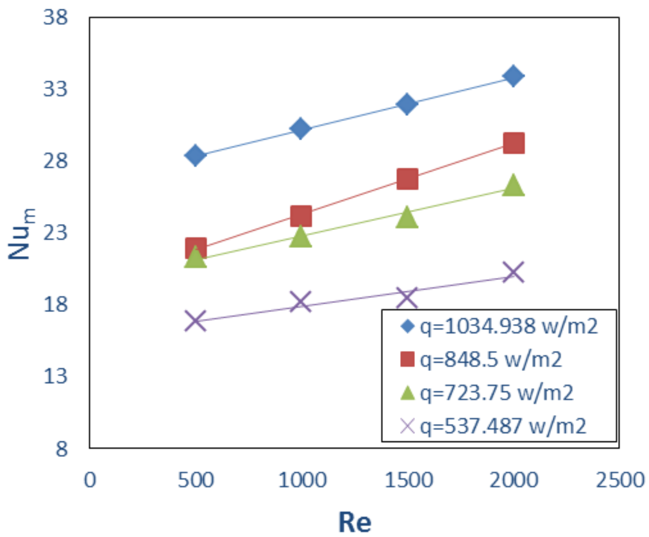



Figure (12): Effect of heat flux on average Nusselt number.

**Correlations of Average Heat Transfer Data**

Figure (13) shows the relationship of ($Nu_m$) against (Ra/Re) for Reynolds number ranges of (500 - 2000) and average Rayleigh number are ($2.638x10^5$ - $3.907x10^5$). Nusselt numbers resulted from all the experiments performed in this work, were correlated in terms of the relevant parameters with an empirical equation. As explained in the previous sections, Ra and Re are the main dimensionless numbers which account for considerable contribution of forced and free convections in this studyThe empirical Nusselt number can be deduced as follows;

$$Nu_m=12.887\ (Ra/Re)^{0.0819} \qquad \dots (17)$$

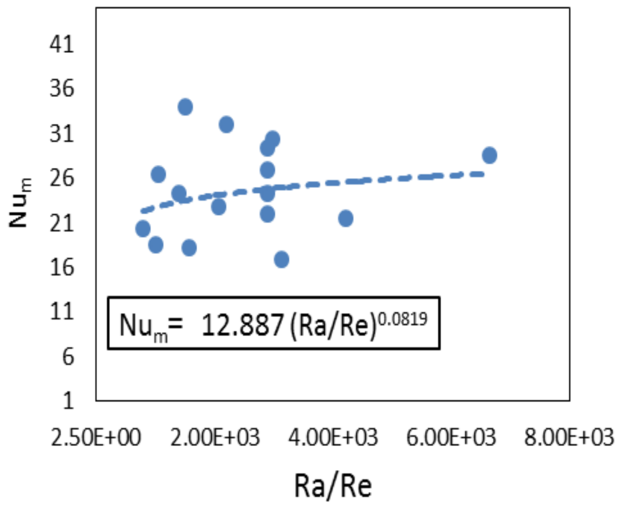



Figure (13) : Average Nusselt number Versus (Ra/Re).

## V. Conclusions

1. The heat transfer process improves (higher values of local Nusselt number and lower values of temperature) as:

a. Heat flux increases (dominant natural convection).
b. Reynolds number increases (dominant forced convection).
2. The secondary currents are week at the entrance region and become strong downstream.
3. An empirical equation for the average Nusselt number as a function of (Ra) and (Re) has been deduced with a wide ranges of Reynolds number and heat flux.

## VI. NOMENCLATURE

A Cylinder cross-sectional area $m^2$
$A_2$ Outer surface area of cylinder $m^2$
$D_h$ Hydraulic diameter mm
g Gravitational acceleration $m/s^2$
h Heat transfer coefficient $W/m^2.°C$
I Electrical current Am
K Thermal conductivity W/m.°C
L Cylinder length m
Q Convection heat flux $W/m^2$
$Q_{cond}$ Conduction heat loss W
$Q_t$ Total heat input W
$T_s$ Semicircular Cylinder surface temperature °C
V Voltage Volt
$\dot{V}$ Volumetric flow rate $m^3/s$
X Axial coordinate m

**Creak Symbols**

| | | |
|---|---|---|
| α | Cylinder inclined angle | degree |
| μ | Dynamic viscosity | kg/m.s |
| ν | Kinematics viscosity | $m^2/s$ |
| ρ | Air density | $kg/m^3$ |
| $\rho_i$ | Air density at cylinder entrance | $kg/m^3$ |
| β | Thermal expansion | 1/K |

**Dimensionless Group**

| | | |
|---|---|---|
| Gr | Grashof number | $\frac{g\beta q r_1^4}{k\nu^2}$ |
| $G_Z$ | Graetz number | $Re.Pr.D_h / x$ |
| Nu | Nusselt number | $h.D_h/K$ |
| $Nu_x$ | Local Nusselt number | $\frac{qD_h}{k(t_{sx}-t_{m_x})}$ |
| Pr | Prandtal number | $\frac{\mu C_p}{k}$ |
| Ra | Rayligh number | Gr.Pr |
| Re | Reynolds number | $u_i D_h / \nu$ |
| Ri | Richardson number | $Gr/\mathrm{Re}^2$ |
| ZZ | Inverse Graetz number | $x/Re.Pr.D_h$ |

**Subscript**

b Bulk
f Film
i Entrance
m Mean
x Local

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