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# LAMINAR AIDING COMBINED CONVECTION IN THE DEVELOPING REGION OF INCLINED CYLINDER

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## ABSTRACT

The influence of natural convection due to buoyancy on the laminar upwards air flow in a uniformly heated inclined circular cylinder has been experimentally studied. The investigation covered a wide range of Reynolds number (450≤Re≤2008), heat flux (95 ≤ q ≤ 898) W/m$^2$, Rayligh number ($1.1132\times10^5 \le Ra \le 3.6982\times10^5$), with different angles of cylinder inclination α=0° (Horizontal), α=30°, α=60° (Inclined) and α=90° (Vertical).Results show that the heat transfer process improves as the angle of inclination moves from vertical to horizontal position. A general empirical equation for average Nusselt number $Nu_m$ as a function of Rayligh number Ra, Reynolds number Re and angle of inclination α was obtained.

## الخلاصة

أجريت دراسة عملية لبحث تأثير الحمل الحر على جريان الهواء الطباقي داخل أسطوانة دائرية مائلة مسخنة تسخين منـتظم. شملت الدراسة مدى رقم رينولدز (450≤ Re ≤2008) ، رقـم رايليـه ($1.1132\times10^5 \le Ra \le 3.6982\times10^5$) ولفـيض حراري W/m$^2$ (95 ≤ q ≤898), وزوايا ميل (عمودي °90 ، °60 ، °30 ،أفقي °α=0). بينت النتائج بـأن عمليـة انتقـال الحرارة تتحسن كلما تغيرت زاوية الميلان من الوضع العمودي الى الوضع الأفقي. من النتائج العملية تـم وضـع معادلـة تجريبية عامة لمعدل رقم نسلت كدالة لرقم رينولدز Re ورقم رايلي Ra ، وزواية الميلان α.

**Keywords: Free and Forced, Convection, Cylinder.**

## INTRODUCTION

The convection heat transfer coefficient and fluid flow characteristics of duct flows are often affected by the presence of gravity, particularly at low or moderate flow rates. The orientation of the duct can have a considerable influence on the velocity and temperature profiles and the associated heat transfer and friction coefficients in the duct. For horizontal cylinder, the buoyancy forces are perpendicular to the main flow direction and give rise to secondary currents in the cross section. For vertical cylinder, the gravity forces are in the main flow direction, and axial symmetry is preserved since there is no secondary flow in the cross-section. In inclined cylinder, however, buoyancy forces act in both the main flow and the cross-stream directions. In practice, this situation is commonly encountered in heat exchanger equipment and in solar collectors. Because of important practical application, laminar combined free and forced convection in horizontal, inclined, and vertical circular duct had been studied theoretically and experimentally by many authors such as; Eckert and Diaguila [1953]; Jackson, Harrison and Boteler [1958]; Mc comas and Eckert [1966]; Zeldin and Schmidt [1972]; Morcos and Bergles [1975]; Pascal and Ralph [1985]; Choudhury and Patankar[1988]**;** Wang, Tsuji, and Nagano[1994]; etc. Akeel [1999], performed experiments to study the local heat transfer by mixed free and forced convection to a simultaneously developing air flow in a vertical and horizontal cylinder for L/D=30 . The results demonstrate an increase in the Nusselt number values along the axial distance as the heat flux increases and as the cylinder moves from vertical to the horizontal position.

Busedra and Soliman [2000], performed an experimental investigation of laminar water flow mixed convection in a uniformly heated inclined semicircular duct under buoyancy assisted and opposed conditions within $\pm 20^{\circ}$. The experiment was designed for determining the effect of inclination on the wall temperature, and local fully-developed Nusselt numbers at three Reynolds numbers 500, 1000, and 1500 and a wide range of Grashof numbers. They found that for the upward inclinations, the value of Nusselt number increases with Grashof number and the inclination angle up to $20^{\circ}$, while the effect of Reynolds Number was found to be small. For the downward inclination, Reynolds number has a strong effect on Nusselt number and the manner by which it varies with Grashof number. The purpose of this paper is to present an experimental study of the simultaneously developing laminar air flow and heat transfer in an inclined uniformly heated circular cylinder in which the heating from the cylinder wall leads to significant buoyancy effects in the upward flow. An empirical equation describes the average Nusselt number as a function of Rayleigh number, Reynolds number, and angle of inclination of cylinder has deduced.

## EXPERIMENTAL APPARATUS AND PROCEDURE

The experimental apparatus shown in **Fig.(1)** consists essentially of cylinder as an open air loop test section, mounted on an iron frame (I) which can be rotated around a horizontal spindle. The inclination angle of the cylinder can thus be adjusted as required. An open air circuit was used which included a centrifugal fan (B), orifice plate section (C), settling chamber (F), test section and a flexible hose (E). The air which is driven by a centrifugal fan can be regulated accurately by using a control valve and enters the orifice pipe section and then settling chamber through a flexible hose (E). 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 (G). The air then passed through 1.2 m long test section.

A symmetric flow and a uniform velocity profile produced by a well designed Teflon bell mouth (H) which is fitted at the entrance of aluminum cylinder (N) and bolted in the other side inside the settling chamber (F). Another Teflon piece (H) represents the cylinder exit and has the same dimensions as the inlet piece. The Teflon was chosen because its low thermal conductivity in order to reduce the heat losses from the aluminum cylinder ends. The inlet air temperature was measured by one thermocouple located in the settling chamber (F) while the outlet bulk air temperature was measured by two thermocouples located in the test section exit (R). The local bulk air temperature was calculated by using a straight line interpolation between the measured inlet and outlet bulk air temperature. The test section consists of 7.5 mm wall thickness, 59.3 mm outside diameter and 1.2m long aluminum cylinder. The cylinder is heated electrically using an electrical heater which consists of 1 mm in diameter and 60 m in length nickel-chrome wire (L) electrically isolated by ceramic beads, wounds uniformly as a coil with 10 mm pitch. The outside of the test section was then thermally insulated, covered with asbestos rope layer and fiber glass of 60 mm and 5.7 mm as thickness , respectively. To enable the calculation of heat loss through the lagging to be carried out, six thermocouples are inserted in the lagging as two thermocouples at three locations along the heated section 390 mm apart. Using the average measured temperature drop and thermal conductivity of lagging the heat losses through lagging can be calculated. The cylinder surface temperatures were measured by eighteen asbestos sheath thermocouple (type K). All the thermocouple wires and heater terminals were taken out the test section. To determine the heat loss from the test section ends, two thermocouples were fixed in each Teflon piece. Knowing the distance between these thermocouples and the thermal conductivity of the Teflon, the heat ends loss could thus be calculated. The thermocouple circuit consists of a digital electronic thermometer (type TM-200, serial no. 13528), connected in parallel to the thermocouples by leads through a selector switches. An orifice plate British Standard Unit (BSU) of diameter of (50 mm) and discharge coefficient of (0.6099) was used to compute the flow rate through the cylinder, by using the following equation.

$$\dot{V} = C_d.(\pi D_o^2/4).\sqrt{2g\Delta h}$$

Where:

$\dot{V}$ = flow rate = $mm^3/sec$

$C_d$ = discharge coefficient = 0.6099

$D_o$ = diameter of orifice =50 mm

The following steps were followed throughout the experimental procedure:

1. Adjust the required inclination angle of the cylinder.
2. Switch on the centrifugal fan to circulate the air, through the open loop. A regulating valve was used for adjusting the required mass flow rate.
3. Switch on the electrical heater then adjust the input power to give the required heat flux.
4. The apparatus was left at least three hours to establish steady state condition. The thermocouples readings were taken every half an hour by means of the digital electronic thermometer until the reading became constant, a final reading was recorded. The input power to the heater could be increased to cover another run in a shorter period of time and to obtain steady state conditions for next heat flux and same Reynolds number. Subsequent runs for other Reynolds number and cylinder inclination angle ranges were performed in the same previous procedure.
5. During each test run , the following readings were recorded:
   - **a.** The angle of inclination of the cylinder in degree.
   - **b**. The reading of the manometer (air flow rate) in mm $H_2o$.
   - **c**. The readings of the thermocouples in ˚C.
   - **d**. The heater current in Amperes.
   - **e**. The heater voltage in volts.

## EXPERIMENTAL ANALAYSIS AND CALCULATION

Simplified steps were used to analyze the heat transfer process for the air flow in a cylinder subjected to a uniform heat flux.

**a-** The total input power supplied to the cylinder is obtained by,

$$Q_t=V \quad (1)$$

**b-**The convection and radiation heat transferred from the cylinder is given by :

$$Q_{cr}=Q_t-Q_{cond} \quad (2)$$

where $Q_{cond}$ is the conduction heat loss expressed by,

$$Q_{cond} = \frac{\Delta T_{oi}}{\frac{\ln \frac{r_o}{r_i}}{2\pi k_a L}} \quad (3)$$

where:

$\Delta T_{oi}=T_o-T_i$

$T_o$, is the average outer asbestos rope layer lagging surface temperature.

$T_i$, is the average cylinder surface temperature.

**c-**The convection and radiation heat flux can be represented by:

$$q_{cr}=Q_{cr}/A_1 \quad (4)$$

where:

$A_1=2\pi\, r_i\, L$

**d-**The local radiation heat flux can be calculated as follows:

$$q_r = F_{1-2}\,\varepsilon\,\sigma\left[\left((T_s)_z+273\right)^4-\left(\overline{(T_s)_z}+273\right)^4\right] \quad (5)$$

where:

$(T_s)_z, \overline{(T_s)_z}$ =are local and average temperature of cylinder wall, respectively .

$\sigma$ = Steavan Boltzman constant $=5.66\times10^{-8}$ $W/m^{\circ}K^4$

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

$F_{1-2}\approx 1$ = shape factor.

Then the convection heat flux at any position is:

$$q=q_{cr}-q_r \quad (6)$$

Since the radiation heat flux is very small and can be neglected, then

$q_{cr}\approx q$

**e-**The local heat transfer coefficient can be obtained by,

$$h_z=\frac{q}{(T_s)_z-(T_b)_z} \quad (7)$$

and the local mean film temperature:

$$(T_f)_z=\frac{(T_s)_z+(T_b)_z}{2} \quad (8)$$

where $(T_b)_z$, Local bulk air temperature.

**f-**The local Nusselt number ($Nu_z$) then can be determined as:

$$Nu_z=\frac{h_z\ D_h}{k} \quad (9)$$

And the average values of Nusselt number $Nu_m$ can be calculated as follows:

$$Nu_m = \frac{1}{L}\int_o^L Nu_z dz \tag{10}$$

The average values of the other parameters can be calculated based on calculation of average cylinder surface temperature and average bulk air temperature as follows:

$$\overline{T_s} = \frac{1}{L}\int_{z=0}^{z=L} (T_s)_z \quad dz \tag{11}$$

$$\overline{T_b} = \frac{1}{L}\int_{z=0}^{z=L} (T_b)_z \quad dz \tag{12}$$

and the average mean film temperature is:

$$\overline{T_f} = \frac{\overline{T_s} + \overline{T_b}}{2} \tag{13}$$

All the air physical properties ρ, μ, ν, and k were evaluated at the average mean film temperature ($\overline{T_f}$).

## RESULTS AND DISCUSSION

### Temperature Variation

Generally, the variation of surface temperature along the surface cylinder may be affected by many variables such as heat flux, Reynolds number, and cylinder inclination angles.

The temperature variation in the horizontal position is plotted for selected runs in **Figs.(2, 3, 4 and 5)**. **Fig.(2)** shows the variation of the surface temperature along cylinder for Re=450 and various values of heat flux. The surface temperature increases at the cylinders entrance and attains a maximum point after which the surface temperature begins to decrease. The rate of surface temperature rises at early stage is directly proportional to the wall heat flux. This can be attributed to the increasing of the thermal boundary layer faster due to buoyancy effect as the heat flux increases for the same Reynolds number. The point of maximum temperature on the curve represents actually the starting of thermal boundary layer fully developed. The region before this point is called the entrance of cylinder, The same behavior seems in the **Fig.(3)** for Reynolds number 2008, but with lower temperature because of dominant forced convection in the heat transfer process.

**Figs.(4 & 5)** show the effect of Reynolds number variation on the cylinder surface temperature for lower and higher heat flux (q=145 $W/m^2$,q=898 $W/m^2$); respectively. It is obvious that the increasing of Reynolds number reduces the surface temperature as heat flux kept constant. It is necessary to mention that as heat flux increases the cylinder surface temperature increases because the free convection is the dominating factor in the heat transfer process. The axial temperature variation trend of cylinder surface for vertical and inclined cylinder (α=30°, 60° and 90°) is the same as that obtained in the horizontal position. The extent of mixed convection depends on the magnitude of the heat flux and Reynolds number for the same angle of inclination. When heat flux and Reynolds number are kept constant, the extent of local mixing due to the buoyancy effect in horizontal cylinder is larger than other cylinder angles of inclination. It has been proved that for the same condition of flow rate and input heat flux, the surface temperature variation along the cylinder decreases as the angle of inclination changes from vertical to horizontal position.

### Angle of Inclination Effect on the Temperature Distribution

The variations of cylinder surface temperature along the axial distance for the same heat flux and Reynolds number, and for different angles of inclination (α=0°, 30°, 60°, and 90°) are shown in **Figs. (6 to 9)**. Figures show a reduction in surface temperature with mean approximated value 11% as the angle of cylinder inclination moves from vertical to horizontal position; this can be attributed to the large buoyancy effect in a horizontal cylinder compared with the other angles of cylinder inclination.

**Fig.(6)** shows the influence of the cylinder orientation on the cylinder surface temperature for (q=115$W/m^2$ & Re=450, gives Ri=0.91). It is observed that when the angles change from $60^o$ to $90^o$ at specific axial distance (x=0.37 m) the cylinder surface temperature for the vertical position is lower than that in the inclined position (α=$60^o$). Then beyond this location a reverse trend takes place and the cylinder surface temperature for the vertical position becomes higher than that in the inclined position (α=$60^o$). This behavior can be explained that at cylinder entrance, the effect of free convection is small and forced convection is dominant in the heat transfer process causes the cylinder surface temperature in vertical position less than that in the inclined position (α=$60^o$), but after a certain axial distance a significant reduction in the cylinder surface temperature exists as the cylinder orientation changes from the vertical to the inclined position ($\alpha=60^o$) due to the effect of free convection which begins to dominant factor in the heat transfer process and reduces the cylinder surface temperature. If the Reynolds number is increased to 2008 (gives Ri=0.051)as shown in **Fig(7)** , the behavior of surface temperature distribution along the axial distance is relatively as same as **Fig(6)**.

**Fig(8**) shows the influence of the cylinder orientation on the cylinder surface temperature for ($q=816W/m^2$ & Re=450, gives Ri=2.1). It is observed that when the angle changes from ($\alpha=0^o$ to 30°) at a specific axial distance (x=0.72m) at the cylinder entrance the cylinder surface temperature for the inclined position ($\alpha=30^o$) is lower than that in the horizontal position ($\alpha=0^o$). If the Reynolds number is increased to 2008 (gives Ri=0.11) as shown in **Fig(9)** , the behavior of surface temperature distribution along the axial distance is relatively as same as **Fig(8).**

## Local Nusselt Number ($Nu_x$)

The variation of local Nusselt number ($Nu_x$) with a logarithmic dimensionless axial distance (inverse Graetz number, $zz=\frac{x/D_h}{Pr\,Re}$ ), for horizontal position are plotted, for a selected runs, in **Figs.(10 to 12)**.

The effect of heat flux on the $Nu_x$ for Re=450 is shown in **Fig.(10)**. It is clear from this figure that at the higher heat flux, the results of the local Nusselt number are higher than the results of lower heat flux. This may be attributed to the secondary flow superimposed on the forced flow effect which increases as the heat flux increases leading to higher heat transfer coefficient.

The effects of Re on $Nu_x$ variation with zz are shown in **Figs.(11) & (12)** for heat flux equal to 145 $W/m^2$ and 898 $W/m^2$ ; respectively. For constant heat flux, results depicted that the deviation of $Nu_x$ value moves towards the left and increases as the Reynolds number increases because of decreasing of inverse Greatz number zz . This situation reveals the domination of forced convection on the heat transfer process which improves as Reynolds number increases. The behavior and trend of the variation of local Nusselt number ($Nu_x$) with a logarithmic dimensionless axial distance, for inclined and vertical position ($\alpha$=30°, 60° and 90°) is the same as that obtained in the horizontal position.

## Angle of Inclination Effect on the Local Nusselt Number

The local Nusselt number variation with zz for different angles of inclination and for ($q=115$ $W/m^2$ and Re=450, gives Ri=0.91) and ($q=816$ $W/m^2$ and Re=2008, gives Ri=0.1) are shown in **Fig.(13) & Fig.(14)**; respectively. This figure indicates that for all zz values, the $Nu_x$ value for horizontal position is higher than that for inclined and vertical position (with mean approximated value 5% between $\alpha=0^o$ and $\alpha=90^o$ for **Fig.(13)** and approximated value 10% for **Fig.(14)**. As explained before the dominant free convection, for horizontal cylinder creates a longitudinal vortex along the cylinder which its intensity reduces as the angle of inclination moves from horizontal to vertical position leads to reducing of heat transfer coefficient.

**Fig. (15)** shows the influence of the cylinder orientation on the $Nu_x$ for Re=2008 and $q=115$ $W/m^2$ (gives Ri=0.051). It is observed that when the angles changes from ($\alpha=0^o$ to $30^o$) at the specific zz value (0.001), the $Nu_x$ value for the inclined position ($\alpha=30^o$) is higher than that in the horizontal position ($\alpha=0^o$). Then beyond this zz a reverse trend takes place and the values of $Nu_x$ for the inclined position ($\alpha=30^o$) become lower than that in the horizontal position ($\alpha=0^o$) and closer to that ($\alpha=0^o$). This behavior can be attributed to the small buoyancy effect at cylinder entrance and the forced convection is being the dominating factor in the heat transfer process. But in down stream the secondary flow becomes more effective which improves the heat transfer process. Hence, the heat transfer appears to be higher at horizontal position and its effect is reduced as cylinder orientation moves towards the vertical position.

In horizontal cylinder, the effect of the secondary flow is high; hence at low Reynolds number and high heat flux, situation makes the free convection predominant. Therefore two Cells will be grown about each side of the vertical centre line of cylinder. The cellular motion behaves so as to reduce the temperature difference between the cylinder surface and the air flow, leads to increase the growth of the hydrodynamic and thermal boundary layers along the cylinder and causes an improvement in the heat transfer coefficient. But at low heat flux and high Reynolds number the situation makes forced convection predominant and decreasing of vortex strength leads to decrease the temperature difference between the heated surface and the air, hence, the $Nu_x$ values become close to the vertical cylinder values for the same conditions.

### Correlation of Average Heat Transfer Data

The values of the average Nusselt number ($Nu_m$) for horizontal position ($\alpha$=0$^o$), inclined position for ($\alpha$=30$^o$, 60$^o$), and vertical position ($\alpha$=90$^o$) are plotted in **Figs.(16 to 19)** in the form of log($Nu_m$) against log(Ra/Re) for the range of Re from 450 to 2008, and Ra from $1.1132\times10^5$ to $3.6982\times10^5$. These values of Reynolds number and Rayligh number give a range of Richardson number between 0.0395 (Forced convection dominating) and 2.6 (mixed convection with stronger natural convection than forced convection)

$\alpha$=90$^o$(vertical) $Nu_m=2.993\,(Ra/Re)^{-0.7941}$ (14)

$\alpha$=60$^o$ $Nu_m=2.831\,(Ra/Re)^{-0.6537}$ (15)

$\alpha$=30$^o$ $Nu_m=2.982\,(Ra/Re)^{-0.7823}$ (16)

$\alpha$=0$^o$ (horizontal) $Nu_m=2.962\,(Ra/Re)^{-0.6369}$ (17)

The heat transfer equation at any angle of inclination was deduced in the following from:

$$(Nu_m)_{inc.}=7.7515\ .(Ra)^{-0.639}\ .(Re)^{0.6411}\ .(\alpha)^{-0.0137} \quad (18)$$

where: $\alpha$ is measured in degree

## CONCLUSIONS

1. Generally, the buoyancy effect is very weak at the entrance.
2. The heat transfer process improves as the heat flux increases where Reynolds number is kept constant and vice versa.
3. The horizontal position is the better position to achieve higher heat transfer coefficients which decrease as the cylinder position deviates from horizontal to vertical position.
4. Four empirical equations has been deduced for average Nusselt number as a function to Reynolds number and Rayligh number for angles of inclination; $\alpha$=0$^o$ (horizontal), 30$^o$, 60$^o$, and 90$^o$ (vertical).
5. General empirical equation has been deduced for average Nusselt number at any angle of inclination {i.e., $Nu_m$=f(Re, Ra, $\alpha$)}.

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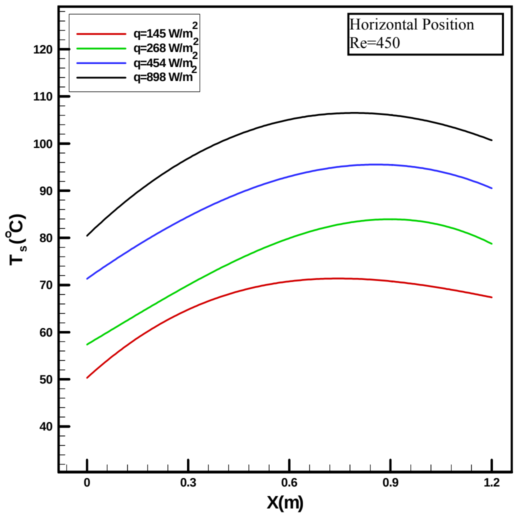


**Fig.(2):** Experimental Variation of the Surface Temperature with the Axial Distance, Re=450, $\alpha$ =0° (Horizontal).

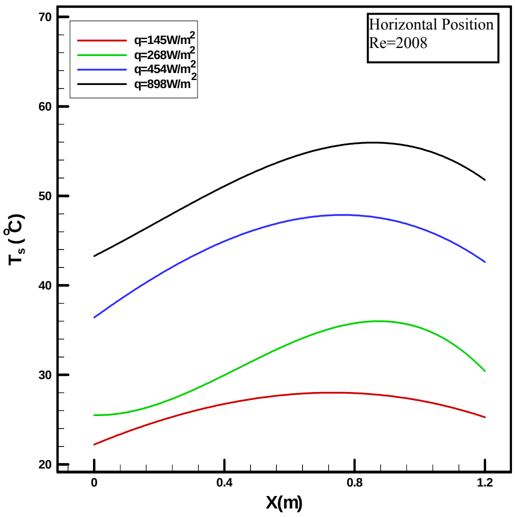


**Fig.(3):** Experimental Variation of the Surface Temperature with the Axial Distance, Re=2008, $\alpha$ =0° (Horizontal).

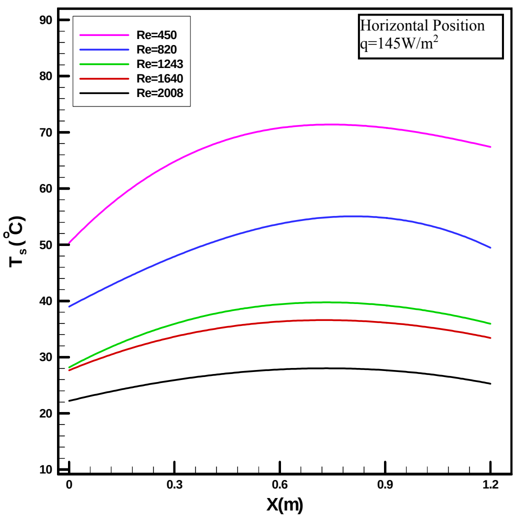


**Fig.(4):** Experimental Variation of the Surface Temperature with the Axial Distance, q=145W/m², $\alpha$ =0° (Horizontal).

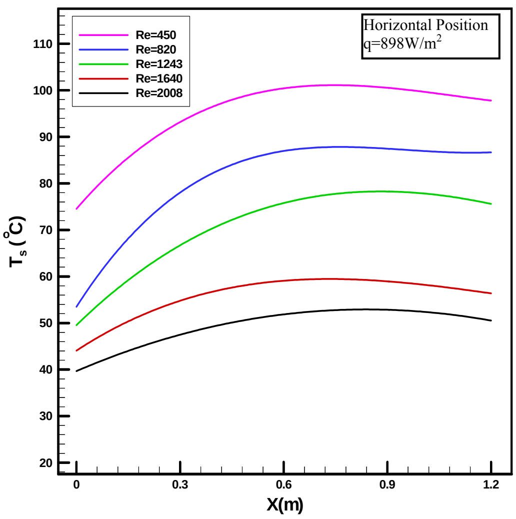


**Fig.(5):** Experimental Variation of the Surface Temperature with the Axial Distance, q=898W/m², $\alpha$ =0° (Horizontal).

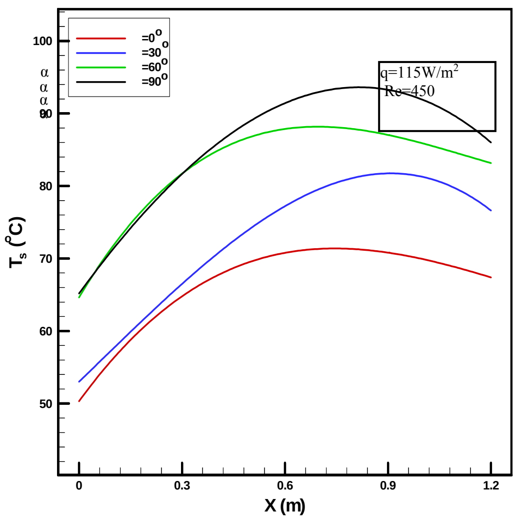


**Fig.(6):** Experimental Variation of the Surface Temperature with the Axial Distance for Various Angles, q=115 W/m², Re=450

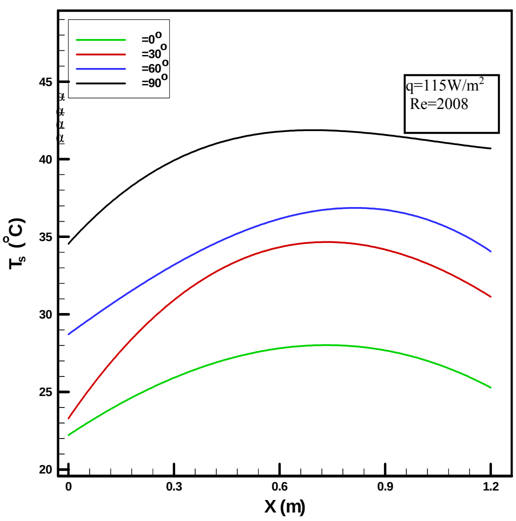


**Fig.(7):** Experimental Variation of the Surface Temperature with the Axial Distance for Various Angles, q=115 W/m², Re=2008

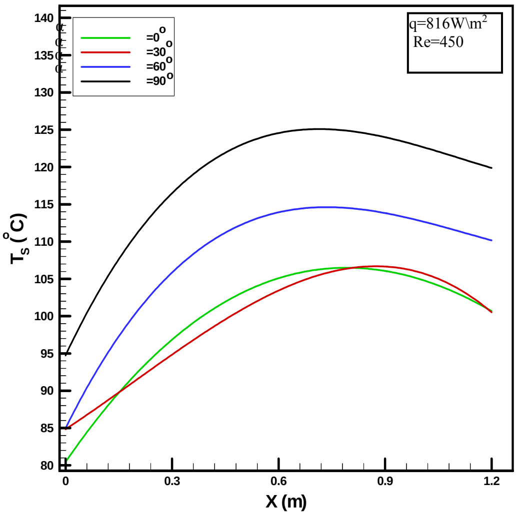


**Fig.(8):** Experimental Variation of the Surface Temperature with the Axial Distance for Various Angles, q=816W/m$^2$, Re=450

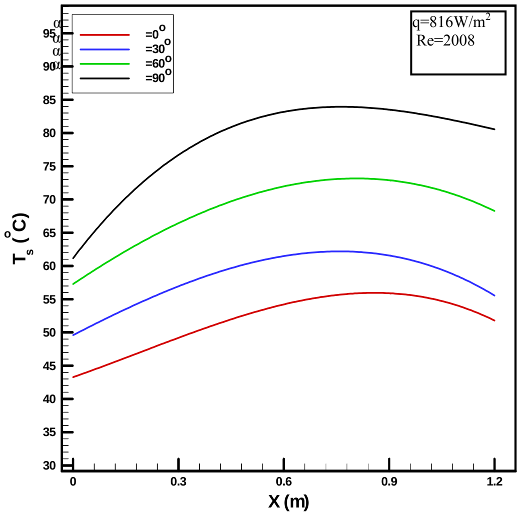


**Fig.(9):** Experimental Variation of the Surface Temperature with the Axial Distance for Various Angles, q=816W/m$^2$, Re=2008

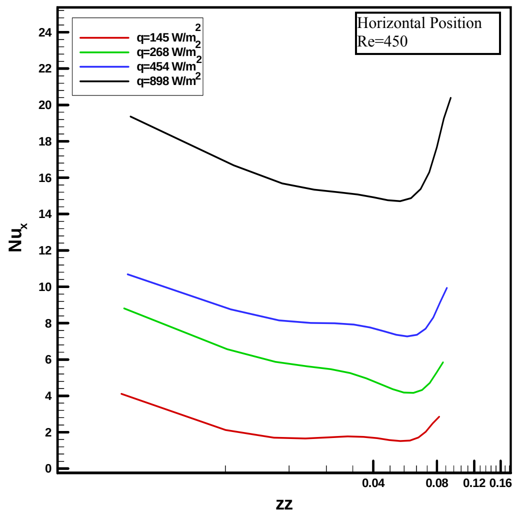


**Fig.(10):** Experimental Local Nusselt number Versus Dimensionless Axial Distance, Re=450, $\alpha$ =0° (Horizontal).

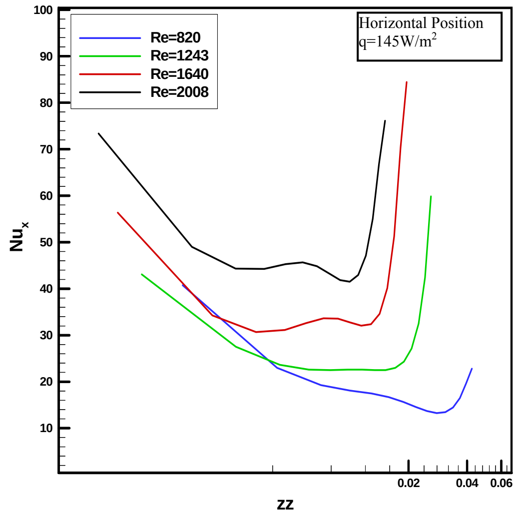


**Fig.(11):** Experimental Local Nusselt number Versus Dimensionless Axial Distance, q=145W/m$^2$, $\alpha$ =0° (Horizontal).

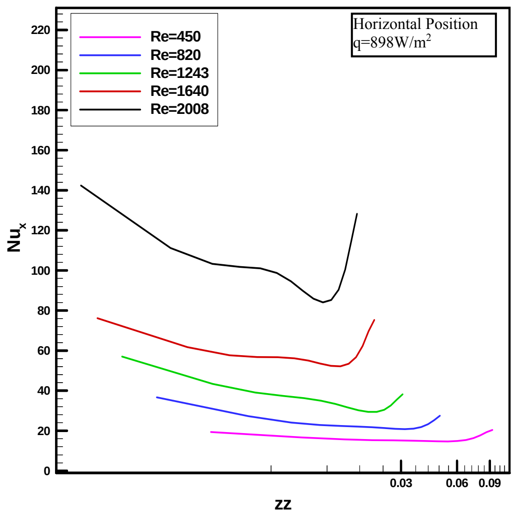


**Fig.(12):** Experimental Local Nusselt number Versus Dimensionless Axial Distance, q=898W/m$^2$, $\alpha$ =0° (Horizontal).

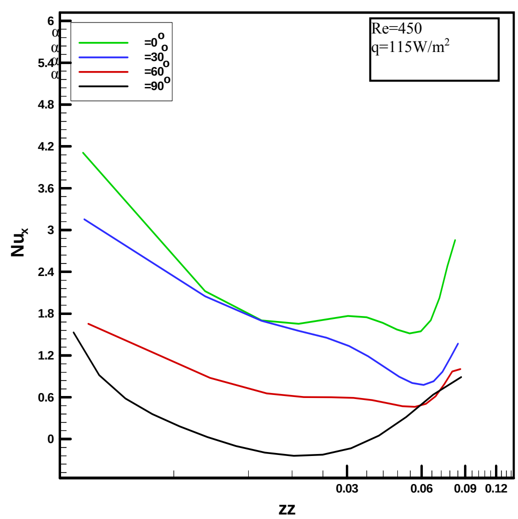


**Fig.(13):** Experimental Local Nusselt number Versus Dimensionless Axial Distance for Various Angles, Re=450, q=115W/m$^2$

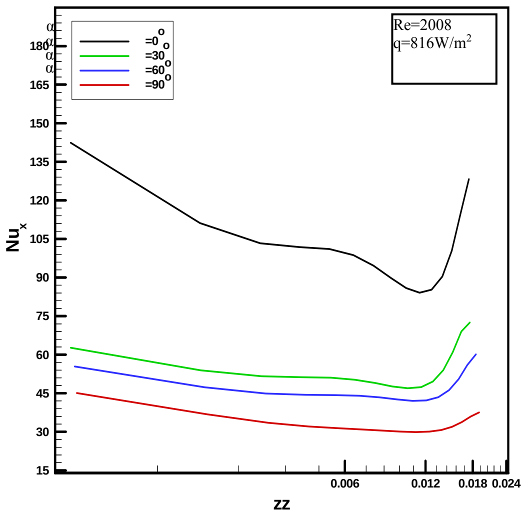


**Fig.(14):** Experimental Local Nusselt number Versus Dimensionless Axial Distance for Various Angles, Re=2008, q=816W/m$^2$

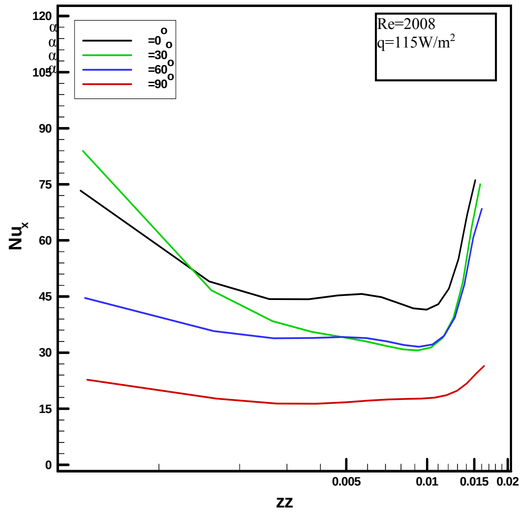


**Fig.(15):** Experimental Local Nusselt number Versus Dimensionless Axial Distance for Various Angles, Re=2008, q=115W/m$^2$

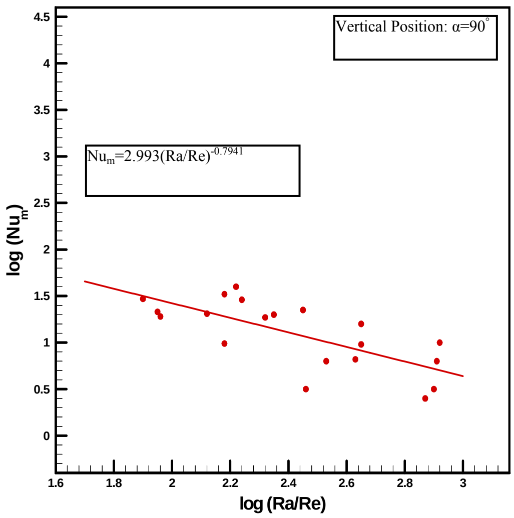


**Fig.(16):** Experimental Average Nusselt number Versus Ra/Re For $\alpha$ = 90° (Vertical Position).

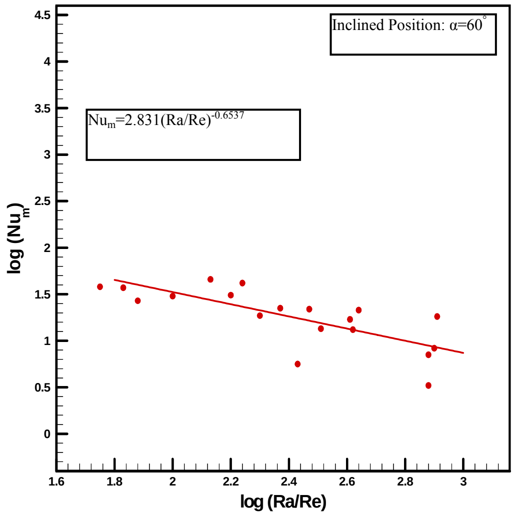


**Fig.(17):** Experimental Average Nusselt number Versus Ra/Re For $\alpha$ = 60° (Inclined Position).

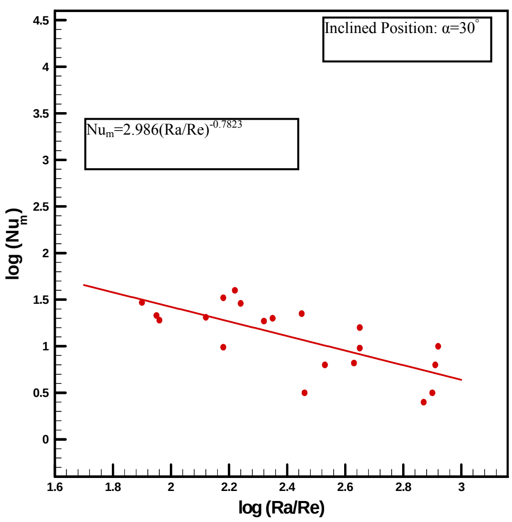


**Fig.(18):** Experimental Average Nusselt number Versus Ra/Re For $\alpha$ = 30° (Inclined Position).

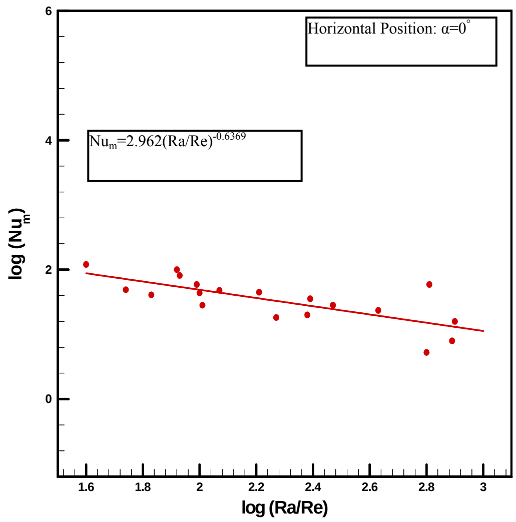


**Fig.(19):** Experimental Average Nusselt number Versus Ra/Re For $\alpha$ = 0° (Horizontal Position).

## NOMENCLATURE

### Latin Symbols:

| Symbol | Description | Unit |
|---|---|---|
| A, | Cylinder cross section area | ($m^2$) |
| $A_1$, | Cylinder surface area | ($m^2$) |
| $C_p$, | Specific heat at constant pressure | (J/Kg.ºC) |
| $D_h$, | Hydraulic diameter | (m) |
| G, | Gravitational acceleration | ($m/s^2$) |
| H, | Coefficient of heat transfer | ($W/m^2$.ºC) |
| I, | Current | (amp) |
| K, | Thermal conductivity | ($W/m^2$.ºC) |
| $K_a$, | Thermal conductivity of asbestos | ($W/m^2$.ºC) |
| L, | Cylinder length | (m) |
| $\dot{V}$, | Volumetric flow rate | ($m^3/s$) |
| Q, | Convection heat flux | ($W/m^2$) |
| $Q_{cond}$, | Conduction heat loss | (W) |
| $Q_t$, | Total heat given | (W) |
| $r_1$, | Inner radius of cylinder | (m) |
| $r_o$, | The distance from center of cylinder to the outer lagging surface | (cm) |
| $r_i$, | The distance from center of cylinder to the beginning lagging (radius of outer cylinder surface) | (cm) |
| $t_b$, | Bulk air temperature | (ºC) |
| $t_f$, | Mean film air temperature | (ºC) |
| $t_i$, | Air temperature at cylinder entrance | (ºC) |
| $t_s$, | Cylinder surface temperature | (ºC) |
| v, | Voltage | (Volt) |
| x, | Axial coordinate | (m) |

### Creak:

| | | |
|---|---|---|
| $\alpha$, | cylinder Inclined angle | (degree) |
| $\mu$, | Dynamic viscosity | (Kg/m.s) |
| $\nu$, | Kinematics viscosity | ($m^2/s$) |
| P, | Air density at any point | ($Kg/m^3$) |

### Subscript:

| | |
|---|---|
| º | Degree |
| x | Local |
| m | Average |

### Dimensionless Gropes:

| | | |
|---|---|---|
| Gr, | Grashof number | $\frac{g\beta(t_s - t_b)D_h^3}{\nu^2}$ |
| Nu, | Nusselt number | $\frac{hD_h}{k}$ |
| Pr, | Prandtal number | $\frac{\mu C_p}{k}$ |
| Ra, | Rayligh number | Gr.Pr |
| Re, | Reynolds number | $\frac{u_i D_h}{\nu}$ |
| Ri | Richardson number | $\frac{Gr}{\mathrm{Re}^2}$ |
| ZZ, | Axial distance | $x/Re.Pr.D_h$ |