Eng. & Tech. Journal, Vol.28, No.15 , 2010

Experimental Study of The Effect of Addition of Calming Section on The Heat Transfer Process In A Uniformly Heated Inclined Tube

Dr. Abdulhassan A. Karamallah*, Dr. Akeel A. Mohammed*, & Nasser F. Hussain**

Received on: 7/2/2010
Accepted on: 3 /6/2010

Abstract.

Experimental research has been conducted for combined convection to study the local and average heat transfer for hydrodynamically fully developed (the value of L/D of calming section was 22.2) and thermally developing laminar air flow in a uniformly heated inclined tube. The study covered a wide range of Richardson number (Ri) from 0.08 to 2.5, Reynolds number from (518 to 2041), and Rayleigh number from (2.4×10^5 to 3.9×10^5), with different angles of inclination {$\alpha = 0^0$ (horizontal), $\pm30^0$, $\pm60^0$, $\pm90^0$ (vertical)} where the minus sign refers to opposing flow and plus sign refers to aiding flow. An empirical correlations were made for the average Nusselt number as a function of Rayleigh number and Reynolds number for all angles of tube inclination and compared with another correlations from previous works. Results show that the heat transfer process for low Richardson number is better than that for high Richardson number and the values of average Nusselt number ($\overline{Nu}$) increase as Peclet number (Pe) increases for the same Rayleigh number.

Key words: Combined Convection, Inclined Tube.

دراسة عملية لتأثير أضافة أنبوب المدخل على عملية أنتقال الحرارة في أنبوب مائل مسخن بصورة منتظمة.

الخلاصة

تم أجراء بحث تجريبي للحمل المختلط لدراسة انتقال الحرارة لجريان هواء طباقي تام التشكيل هيدروديناميكيا (نسبة الطول إلى القطر لأنبوب المدخل هي L/D=22.2) والنامي حراريا في أنبوب مائل مسخن بصورة منتظمة. غطت الدراسة مدى واسع لعدد (Ri) من (0.08 إلى 2.5), عدد (Re) من (518إلى 2041), وعدد (Ra) من 2.4×10^5 إلى 3.9×10^5), مع زوايا ميلان مختلفة {0^0 (أفقي),$\pm30^0$,$\pm60^0$, $\pm90^0$ (عمودي)} حيث الإشارة السالبة تشير إلى الجريان المعاكس والإشارة الموجبة تشير إلى الجريان المساعد. تم التوصل إلى علاقات تجريبية لتغير معدل عدد نسلت بدلالة عدد (Ra) و عدد (Re) لكل زوايا ميلان الأنبوب وتم مقارنتها مع معادلات أخرى من أعمال سابقة. بينت النتائج أن عملية انتقال الحرارة عند عدد (Ri) منخفض تكون أفضل مما هو عليه عند عدد (Ri) عالي وقيم معدل عدد نسلت تزداد بزيادة عدد(Pe) لنفس عدد (Ra).

* Mechanical Engineering Department, University of Technology, Baghdad.
**Electromechanically Engineering Department, University of Technology, Baghdad.

https://doi.org/10.30684/etj.28.15.20
2412-0758/University of Technology-Iraq, Baghdad, Iraq

1. Introduction.

Thermal buoyancy forces play a significant role in forced convection heat transfer when the flow velocity is relatively small and the temperature difference between the surface and the free stream is relatively large. The buoyancy force modifies the flow and the temperature fields and hence the heat transfer rate from the surface. Problems of heat transfer in tubes by combined free and forced convection have been the subject of investigations for many years. Mixed convection occurs in many heat transfer devices, such as nuclear power technology, some aspects of electronics cooling, supercritical boilers, tube heat exchangers, etc. The relative direction between the buoyancy force and the externally forced flow is important. In the case where the fluid is externally forced to flow in the same direction as the buoyancy force, the mode of heat transfer is termed assisting combined forced and natural convection. In the case where the fluid is externally forced to flow in the opposite direction to the buoyancy force, the mode of heat transfer is termed opposing combined forced and natural convection or mixed convection which is characterized by the dimensionless parameter representing the ratio of buoyancy and inertial forces. This parameter is called Richardson number, simply defined as:

$$Ri = \frac{Gr}{\mathrm{Re}^2} \qquad .....(1)$$

Various researchers investigated the effect of mixed convective flows in tubes using experimental methods. **Grassi & Testi [1],** Studied the regime of transitional mixed convection in a uniformly heated horizontal cylindrical pipe. Heat transfer coefficients were measured at 6 cross sections along the heated length, with 60 azimuthal positions being monitored at each section. A single Nusselt number correlation was proposed for the entire heat transfer, best fitting the 5760 experimental points with only 7 parameters. The maximum deviation from the measurements is 17.5%, with a 3% average. **Yousef and Tarasuk [2],** studied the influence of the free convection on forced air flow in the entrance region of an isothermal tube. The L/D ratio was varied from (6 up to 46), Reynolds number ranged from (120 to 1200), and the Grashof number ranged from (0.8×10^4 to 8.7×10^4). The average Nusselt number based on the log-mean temperature difference, ranged from (2 to 25.9), has been obtained. The heat transfer data were correlated according to the influence of the free convection, which was found to have a significant effect at points close to the tube entrance. **Szetela and Sobel [3],** used a Kerosene-type gas turbine fuel flow through a direct-resistance heated tube at uniform heat flux. The data presented were obtained at an entrance Reynolds number of approximately 1500 and led to two peaks in a tube wall temperature near the tube entrance. Results show that at higher entrance Reynolds numbers of 2200 and 3000, the measured wall temperature indicated only a single peak. **Paulo and Genesio [4],** studied the mixed convection heat transfer in an inclined rectangular channel. Three heat sources with finite lengths are flush mounted on the bottom surface of a channel. The Reynolds number ranged from (1 to 1000), Grashof number ranged from (10^3 to 10^5) and inclination angle were (0^0,45^0,90^0). It is observed that the inclination angle has a stronger influence on the flow and heat transfer for Reynolds number equals to 1000, especially when it is between 0^0 and 45^0 because the complicated flow patterns in these angles and the dominated forced convection at Re=1000. **Shannon and Depew [5],** used water initially at ice point as a test fluid in a circular tube with a uniform wall heat flux and fully developed velocity profile at the onset of heating. Reynolds number ranged from (120 to 2300), Grashof number ranged up to (2.5×10^5), and Graetz number ranged from (1.5 to 1000). The Nusselt number was not significantly affected in the thermal entry region, but values which are 2½ times those expected for uniform properties were discovered far down the tube at x/D≈700. **Hussein, et al [6-9]** performed experimentally a study for thermally developing and hydrodynamically fully developed laminar air flow inclined cylinder L/D=30. The tube was under constant wall heat flux boundary conditions. Reynolds number ranged from (400 to 1600) and the heat flux is varied from ($60W/m^2$ to $400W/m^2$). The results had demonstrated that an increase in the Nusselt number values as the heat flux increases and as the angle of cylinder inclination moves from vertical to horizontal position. **Mohammed [11],** performed experimentally 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, Reynolds number ranged from (420 to 1540), and heat flux varied from (62 W/m^2 to 370 W/m^2). The results demonstrate an increase in the Nusselt number values along the cylinder as the heat flux increases and as the cylinder moves from the vertical to the horizontal position.

This study has been presented the effect of calming section which was used to produce hydrodynamically fully developed flow on the heat transfer process and discussed the direction effect of primary flow relative to the secondary motion generated by buoyancy effects and compared the results with previous works to knew the effect of shape of velocity profile (uniform or fully developed) at the entrance region of the test section and the value of L/D for test section on the heat transfer process.

2. Experimental Apparatus

The schematic diagram of all apparatus and experimental rig is shown in Fig.(1-a). The heat transfer test section is a circular tube made from aluminum with dimensions (inner diameter 52.3 mm, outer diameter 59.3 mm, and length 1.2 m). The test section was thermally insulated by using asbestos rope and fiberglass with total thickness of about 65.3 mm to minimize heat losses to the surrounding as shown in Fig.(1-b). A uniform heat flux as a boundary condition was maintained by using an electrical heater wire (nickel-chrome) wound uniformly around the tube as a coil with 10 mm pitch and electrically isolated by ceramic beads to prevent any electrical contact between the test section and electrical heater as shown in Fig.(1-b). An eighteen asbestos sheath thermocouple (type K) were placed along the tube wall to measure the tube surface temperature.

To ensure that the air flows at the entrance of the test section hydrodynamically fully developed, an aluminum tube (calming section) which has the same dimensions of the test section was used for this purpose. The calming section was connected with the test section by teflon connection piece bored with the same inside diameter of the test section and calming section. The teflon material was chosen because of its low thermal conductivity so that the losses from the end of the test section can be reduced.

3. Experimental Procedure

Firstly, the inclination angle of the tube is adjusted as required. Then, the centrifugal fan is switched on to push the air through the open loop while the fan control valve was used for adjusting the required mass flow rate. Then, the electrical heater switched on while the electrical input power adjusted by AC power variac to give the required heat flux. The apparatus is left at least two hours to establish steady state condition, during this time, the thermocouples readings is measured every half an hour by means of the digital electronic thermometer until the reading became constant, then, this reading is recorded as a final reading. During each test run, (angle of inclination of the tube, reading of the manometer, readings of the thermocouples, heater current, and heater voltage are recorded).

4. Data reduction

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

$$Q_t = V'' \times I \qquad \ldots(2)$$

The convection and radiation heat transferred from the tube is :

$$Q_{cr} = Q_t - Q_{cond} \qquad \ldots(3)$$

where Q_{cond} is the total conduction heat loss.

$$q_{cr} = Q_{cr} / A_2 \qquad \ldots(4)$$

where:

A_2=outer surface area of tube $=2\pi r_2 L$.

hence the radiation heat flux is very small and can be neglected

So

$$q_{cr} \approx q = \text{convection heat flux} \qquad \ldots(5)$$

The local heat transfer coefficient can be obtained as:

$$h_x = \frac{q}{(T_s)_x - (T_b)_x} \qquad \ldots(6)$$

$(T_s)_x$ =Local surface temperature, $(T_b)_x$ = Local bulk air temperature.

All the air properties are evaluated at the mean film air temperature [10].

$$(T_f)_x = \frac{(T_s)_x + (T_b)_x}{2} \qquad \ldots(7)$$

where:

$(T_f)_x$ = Local mean film air temperature.

The local Nusselt number (Nu_z) then can be determined as:

$$Nu_x = \frac{h_x * D_h}{k} \qquad \ldots(8)$$

where:

k=thermal conductivity of air = 0.6099 $W/m^2\,°C$

The average values of Nusselt number $\overline{Nu}$ can be calculated as follows:

$$\overline{Nu} = \frac{1}{L}\int_o^L Nu_x\, dz \qquad \ldots(9)$$

5. Results and Discussion

A) Temperature and Local Nusselt number Variations

A total of 140 test runs have been conducted to cover seven tube inclination angles of 0^0, 30^0, 60^0, 90^0, -30^0, -60^0, -90^0. The range of Rayleigh number from (2.4×10^5 to 3.9×10^5) and Reynolds number is varied from (518 to 2041). In this research, the effects of more important parameters on the heat transfer process were taken into consideration such as Ri, Pe, Ra, Re, α, that few figures has been covered.

The surface temperature variation for selected runs is plotted in Fig.2 and Fig.3 for various angles of inclinations and constant Richardson number Ri=1.2 & 0.08 (i.e., constant Reynolds number and heat flux which gives Grashof number); respectively. If the behavior of temperature distribution is studied alone, it has been observed that at the entrance of tube the surface temperature increases from a certain value along tube axis because the thickness of the boundary layer is zero. Then it gradually increases to maximum value when the boundary layer fills the tube. This region is called tube thermal entrance in which the heat transfer results gradually decreases. Beyond this region the surface temperature is decreased due to the laminarization effect in the near wall region (buoyancy effect) and due to tube end losses. It is clear from these two figures that the values of temperature increase as the inclination angles deviates from horizontal to vertical position for both aiding and opposing case. This behavior could be elucidated and attributed to that when heat is transferred through the tube wall at horizontal position (i.e., the primary flow is perpendicular to secondary flow), the density of the air near the wall decreases. Then, the body forces acting due to this density change cause the hotter air to climb upward along the side walls of the tube while the fluid of higher density (lower temperature) moves downward to the tube core then to the lower portion of the tube wall. This related to increase the secondary flow pattern and it has been expected two identical longitudinal vortices have generated which superimposed on the main forced flow as has been observed by lavine et al [14]. These vortices will be weaken as the angle of inclination deviated from horizontal to vertical position in which the vortex diminishes because the primary and secondary flows are parallel with the same or opposite direction. It is necessary to mention from these two figures that the values of temperature for opposing flow are higher than that for aiding flow at the same axial distance and vice versa for the values of local Nusselt number as shown in Fig.4 and Fig.5 which show the variation of local Nusselt number with dimensionless axial distance (inverse Greatz number) for (Ri=0.08 and 1.2) respectively, and for various angles of inclination. This behavior can be attributed to that when the flow is opposing, the direction of main flow opposes the buoyancy force which tends to retard the fluid near the heated wall (the velocities decrease in this region) leading to decrease the natural convection. While in the case of aiding flow, the buoyancy force and the main flow are in the same direction. So, the velocities near the heated wall increase and the free convection assists the main flow in removing heat from the tube surface and improving the heat transfer process. The dotted curve in the last two figures represents the theoretical pure forced convection (TPFC) based on constant property analysis of shah and London [13]. It can be seen also that the values of the local Nusselt number for all angles of inclination are higher than that predicted for (TPFC) for low values of Richardson number as shown in Fig.4, and these values begin to decrease as Richardson number increases as shown in Fig.5 at the same (Gz^{-1}). The observations of these figures can be regarded as conclusions listed later in this item.

B) Average Nusselt number Variations

Variations of the Nusselt number with the increase of the Peclet number at selected Rayleigh number are presented in Fig.6 and Fig.7 for horizontal and vertical position; respectively. It is evidenced that at lower Peclet numbers the effect of Rayleigh number is significant. When the Peclet number is larger than certain value (Pe > 1450) the heat flux data points close to each other for different Rayleigh numbers ($Ra<3.3\times10^5$) which means that the average Nusselt number becomes independent of Ra and forced convection prevails. This observation can also be confirmed by Figs.8 & 9 for horizontal and vertical position; respectively where the variation of the average Nusselt number is plotted against the Rayleigh number at selected Peclet numbers. As shown in these figures, at lower Peclet numbers such as (Pe= 368, 638) the average Nusselt number increases with increasing the Rayleigh number. This suggests the fact that the natural convection is predominated at lower Peclet numbers. On the other hand, when the Rayleigh number is greater than a certain value ($Ra>3.9\times10^5$) the average Nusselt number close to each other for Pe < 895, i.e., the average Nusselt number becomes independent of the Rayleigh number, indicating that the forced convection becomes predominated.

C) Average heat transfer correlation

The values of average Nusselt number are correlated in empirical equation for a horizontal, inclined, and vertical tube (for aiding and opposing flow) and are plotted in Figs.(10-16) in form of $\log\overline{Nu}$ versus $\log\overline{Ra/Re}$ for the range of Re from (518 to 2041) and heat flux from (152 W/m^2 to 812 W/m^2). It was shown that the heat transfer equations for all angles of inclination have this form:

$$\overline{Nu} = A\,(\overline{Ra}\ /\ \mathrm{Re})^{B} \quad \ldots(10)$$

The value of B represents the slope of the straight line. These correlations have been compared with the correlations of Hussein et al [6-9] and Jadoa [12]. The values of A & B for these works and the present work are listed in the table (table A):

The practical importance of these empirical correlations presented in this study will be used for designing of some heat transfer process. It is evident from these figures that the behavior of heat transfer process for the present work agrees well with the work of Jadoa [12] in which the average Nusselt number values decreases as the average value of Ra/Re increases with a higher values for the present work. In the other side, this behavior is reversal in the work of Hussein et al [6-9]. To discuss this difference, a comparison among these three works should be made to know the difference and similarity among them as follows in the table (table B):

As be shown in table (B), the extension in the range of values of Richardson number for Hussein work it compared with the present work leading to reverse the behavior of heat transfer process. In another words, the heat transfer process improves as the average values of Ra/Re increases (i.e., the natural convection is the dominant factor in the heat transfer process) for the high range of Richardson number and as Ra/Re decreases for the low range of Richardson number such as the present work (i.e., the forced convection is the dominant factor in the heat transfer process).

6. Conclusions

Based on the experimental results presented here in, the following conclusions were made:

1. The horizontal position gives lower values of temperature and higher values of local Nusselt number than other angles of inclination.
2. Generally, the heat transfer process in the case of aiding flow is better than that in the case of opposing flow at the same angle of inclination and Richardson number.
3. For Ri=0.08 (very lower than one i.e., mixed convection with dominated forced convection), the values of local Nusselt number for all angles of inclination are very higher

than that predicted for (TPFC) at the same local location which is represented by inverse Greatz number ($G\bar{z}^{1}$).

4. For Ri=1.2 (i.e., strong mixing process), it can be concluded that

a. The values of Nux at $\alpha=\pm60^{0}$ and $=\pm90^{0}$ (vertical), are lower than that predicted for (TPFC), i.e., these angles of inclination give low values of heat transfer coefficient inspite of higher value of Richardson number.

b. The value of Nux at $\alpha=30^{0}$ at the end tip point of thermal entrance length approaches from the asymptotic value of (TPFC), i.e., the point at which the value of Nux becomes constant.

c. The values of Nux upstream at $\alpha=-30^{0}$ are similar to that predicted for (TPFC), then decrease downstream to be lower than that in (TPFC).

d. The local Nusselt number values for $\alpha=0^{0}$ (horizontal position) and $\alpha=30^{0}$ are higher than that predicted for (TPFC) at the same local location ($G\bar{z}^{1}$).

5. At lower Peclet numbers the effect of Rayleigh number is significant. When the Peclet number is larger than certain value the average Nusselt number becomes independent of Ra and forced convection prevails.

6. The heat transfer process improves by using calming section.

7. Richardson number is important factor that indicated the behavior of heat transfer process as follows:

a. For high range of Richardson number, essential player in the heat transfer process is Rayleigh number (i.e., average Nusselt number increases as average Ra/Re increases)

b. For low range of Richardson number , the heat transfer process improves as Reynolds number increases because of dominated forced convection process (i.e., average Nusselt number decreases as average Ra/Re increases).

Nomenclature

A_2 Outer surface area of tube, (m^2).
C_p Specific heat at constant pressure, ($J/Kg.^{0}C$).
D_h Hydraulic diameter, (m).
h Coefficient of heat transfer, ($W/m^{2\,0}C$)..
I Current, (Ampere).
K Thermal conductivity, ($W/m.^{0}C$).
L Tube length, (m).
q Convection heat flux (W/m^2) .
Q_{cond} Conduction heat loss, (W).
Qt Total heat input, (W).
r_1 Inner tube radius, (m).
r_2 Outer tube radius, (m).
t_s Tube surface temperature, (^{0}C).
V'' Heater voltage, (volt).

Greek Symbols

α Cylinder inclined angle, (degree)
μ Dynamic viscosity, (kg/m.s)
ν Kinematics viscosity, (m^2/s)
ρ Air density at any point, (kg/m^3)
β Thermal expansion, (1/K)

Dimensionless Gropes:

Gr Grashof number
Gz Graetz number
Nu_x Local nusselt number
$\overline{Nu}$ Average Nusselt number
Pe Peclet number
Pr Prandtal number
Ra Rayleigh number
Re Reynolds number
Ri Richardson number

References

[1] W. Grassi, D. Testi, "Transitional mixed convection in the entrance region of a horizontal pipe", Department of Energetic"L.Poggi"-University of Pisa, Italy, 2008.

[2] W. W. Yousef and J. D. Tarasuk, "Free convection effects on laminar forced convective heat transfer in a horizontal isothermal tube", J. Heat Transfer, vol.104, pp.145-152, 1982.

[3] E. J. Szetela and D. R. Sobel, "Tube entrance heat transfer with deposit formation", Heat transfer Engineering, vol.4, No.2, pp.47-53, 1983.

[4] Paulo M. Guimaraes and Genesio J. Menon, "Combined free and forced convection in an inclined channel with discrete heat sources", Mecanica Computational vol. XXIII, pp. 2123-2140, 2004.

[5] R. L. Shannon and C. A. Depew, "Combined free and forced laminar convection in a horizontal tube with uniform heat flux", J. Heat Transfer, vol.80, pp.353-357, 1968.

[6] Hussein A. Mohammed and Yasen K. Salman, "Combined natural and forced convection heat transfer for assisting thermally developing flow in a uniformly heated vertical cylinder", J. Heat and mass Transfer, vol.34, pp.474-491, 2007.

[7] Mohammed, h. and Yusaf T " Heat transfer by mixed convection opposing laminar flow from the inside surface of uniformly heated inclined circular tube", Proceeding of ESDA2006, 8th Biennial Conference on Engineering Systems Design and Analysis, July 4-7, 2006, Torino, Italy, pp.1-11.

[8] H.A. Mohammed, and Y.K. Salman, " Experimental investigation of mixed convection heat transfer for thermally developing flow in a horizontal circular cylinder", Applied thermal Engineering vol.27, pp.1522-1533, (2007).

[9] Hussein A. Mohammed and Yasin K. Salman, " Combined convection heat transfer for thermally developing aiding flow in an inclined circular cylinder with constant heat flux" Applied thermal Engineering vol.27, pp.1236-1247, 2007

[10] J. Grimson, "Advance fluid dynamic and heat transfer", Mc Graw-hill, England, (1971).

[11] Mohammed, A. A, "Combined free and forced convection in a circular cylinder", M.Sc. Thesis, University of

Baghdad, College of Engineering, Mechanical Department, 1999.

[12] **Jadoa, A. A,** "An investigation into a laminar combined free and forced convection in a variable cylinder location", M.Sc. Thesis, University of Technology, Mechanical Engineering Department, 2008.

[13] **Shah, R. K and London, A. L,** "Laminar flow forced convection in ducts", Advances in Heat Transfer, supplement1, Academic Press, 1978.

[14] **A.S. Lavine, M.Y. Kim, C.N. Shores,** " Flow reversal in opposing mixed convection flow in inclined pipes", J. heat transfer, ASME transactions vol. III, pp.114-120, 1989.

Table A: Constants in Eq.(1) for the present work and Jadoa and Hussein works

α	Present work		Jadoa work		Hussein et al works	
	A	B	A	B	A	B
0^0	2.9979	-0.6156	2.962	-0.6369	3.19	0.26
30^0	2.9190	-0.6013	2.982	-0.7823	3.25	0.187
60^0	2.7687	-0.5689	2.831	-0.6537	3.55	0.134
90^0	2.5412	-0.4882	2.993	-0.7941	3.71	0.118
-30^0	2.7384	-0.5523	2.131	-0.8055	3.08	0.165
-60^0	2.602	-0.5192	2.891	-0.7224	3.43	0.115
-90^0	2.717	-0.5073	2.855	-0.7773	3.62	0.086

Table B: Comparison the present work with Jadoa and Hussein works.

No.	Present work	Jadoa work	Hussein et al works
1-	Hydrodynamically fully developed flow	Simultaneously developing flow	Thermally developing flow
2-	L/Dcalm=22.2	L/D=22.2	L/D=30
3-	$518 \le Re \le 2041$	$450 \le Re \le 2008$	$400 \le Re \le 1600$
4-	$0.08 \le Ri \le 2.5$	$0.04 \le Ri \le 2.6$	$0.1 \le Ri \le 10$
5-	$2.4\times10^5 \le Ra \le 3.9\times10^5$	$1.1\times10^5 \le Ra \le 3.6\times10^5$	$1.1\times10^5 \le Ra \le 4.5\times10^5$

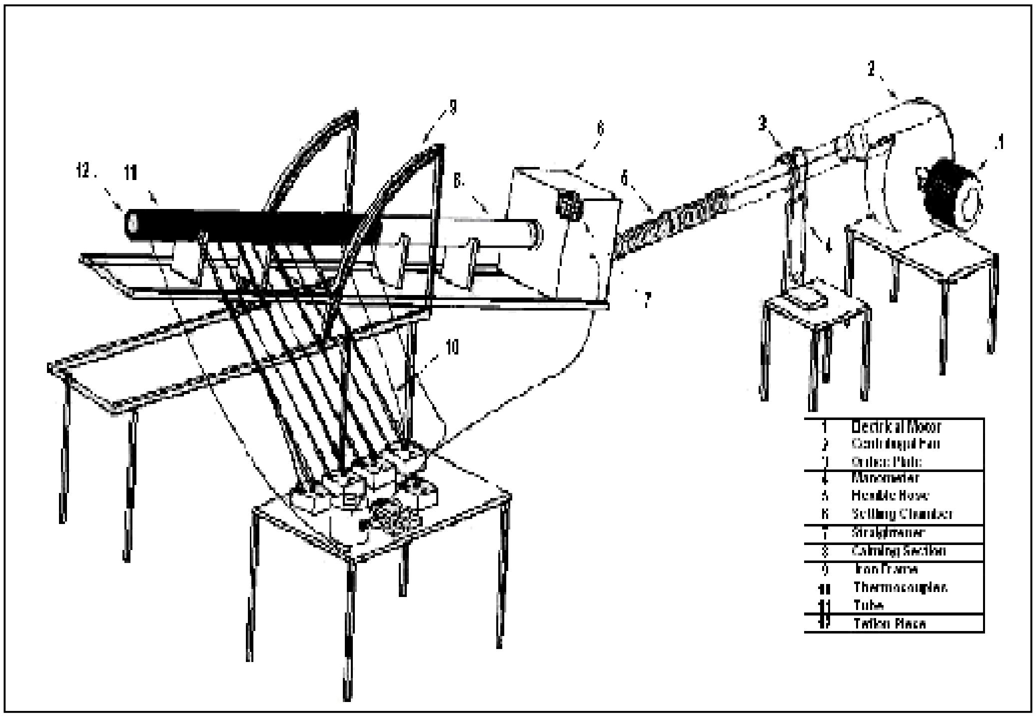


Figure (1-a): Schematic of experimental apparatus.

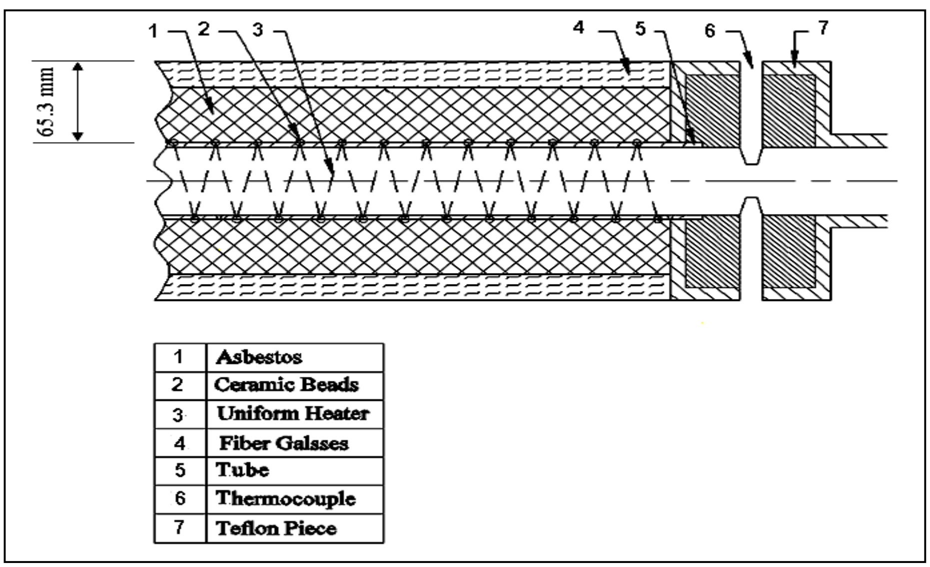


Figure (1-b): Heating system

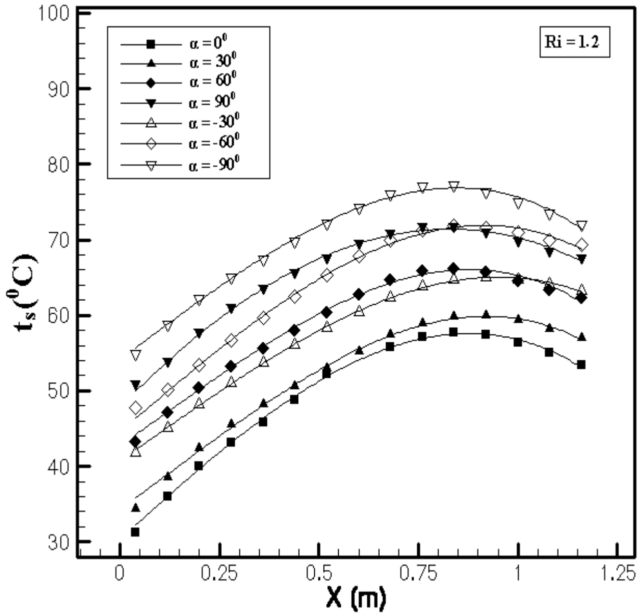


Figure (2): Variations of temperature distribution along x-axis for various angles of inclination, Ri=1.2

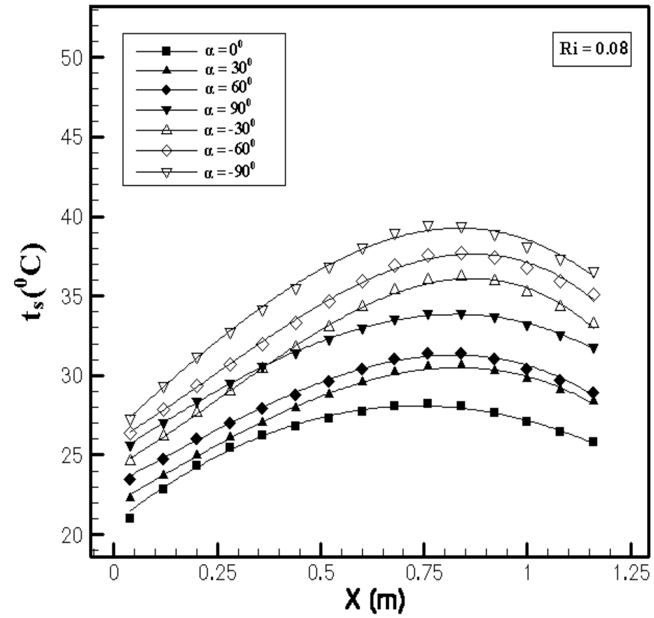


Figure.(3): Variations of temperature distribution along x-axis for various angles of inclination, Ri=0.08

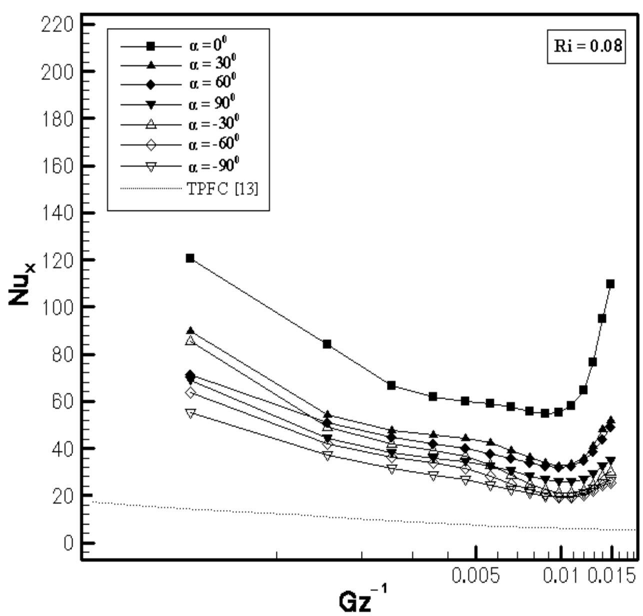


Figure.(4):Local Nusselt number versus inverse Greatz number for various angles of inclination, Ri=0.08

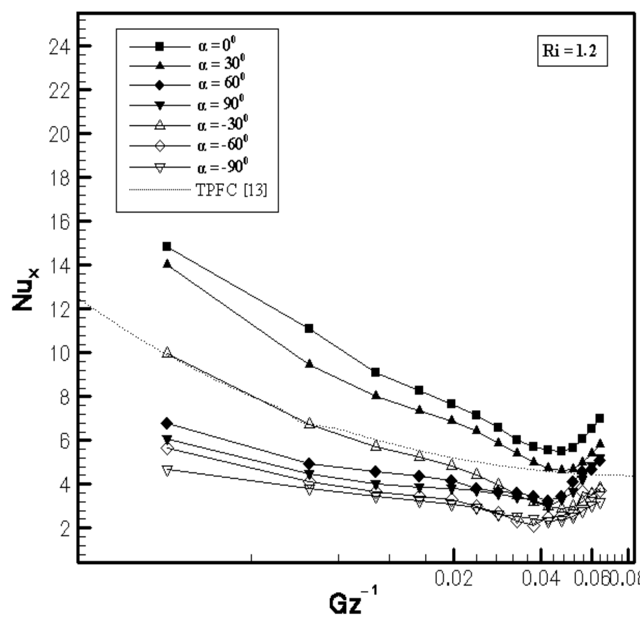
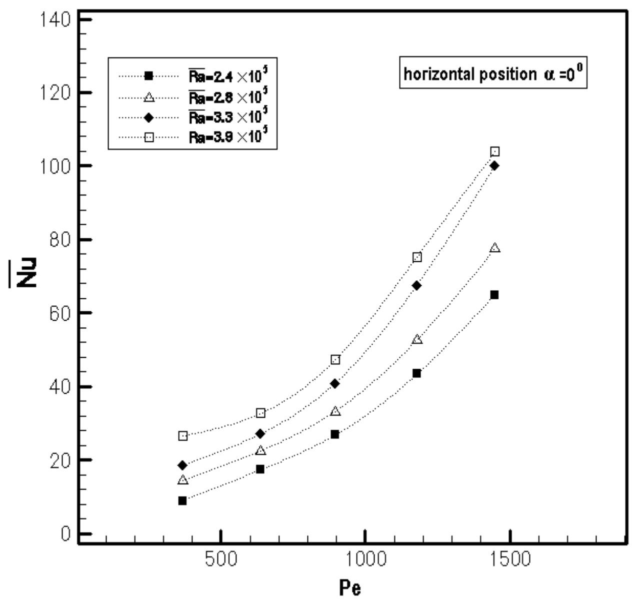


Figure.(5): Local Nusselt number versus inverse Greatz number for various angles of inclination, Ri=1.2



Figure(6): Variations of the average Nusselt number versus Peclet number at selected Rayleigh number, $\alpha=0^0$

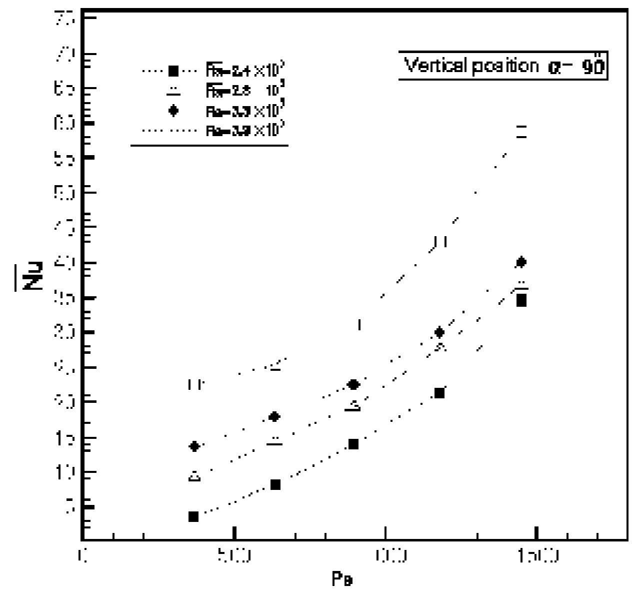
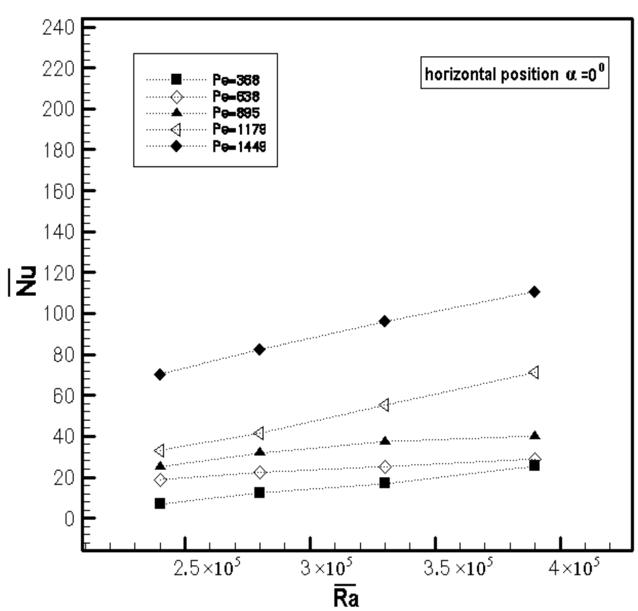
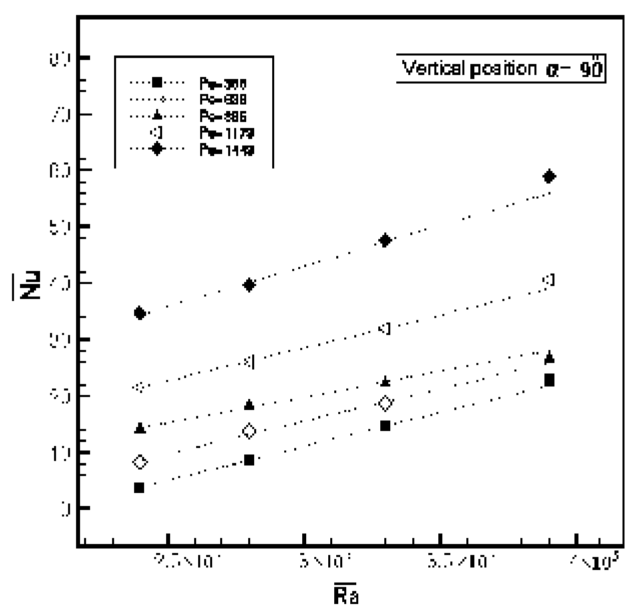


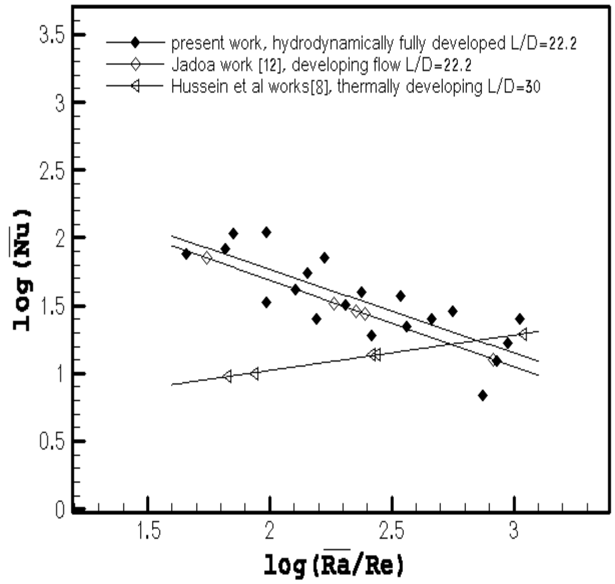
Figure.(7): Variations of the average Nusselt number versus Peclet number at selected Rayleigh number, $\alpha=90^0$



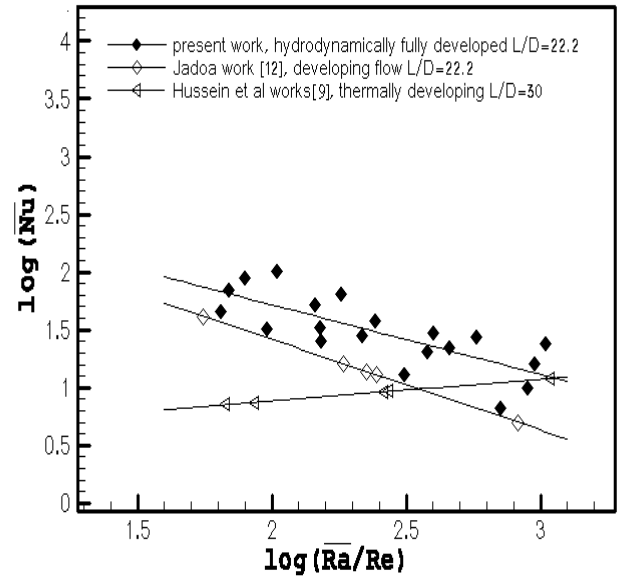


Figure(8): Variations of the average Nusselt number versus Rayleigh number at selected Peclet number, $\alpha=0^0$

Figure.(9): Variations of the average Nusselt number versus Rayleigh number at selected Peclet number, $\alpha=90^0$



Figure(10): Comparison of the average heat transfer results with available literature for α=0^0 (horizontal position)



Figure(11): Comparison of the average heat transfer results with available literature for α=30^0 (Inclined position)

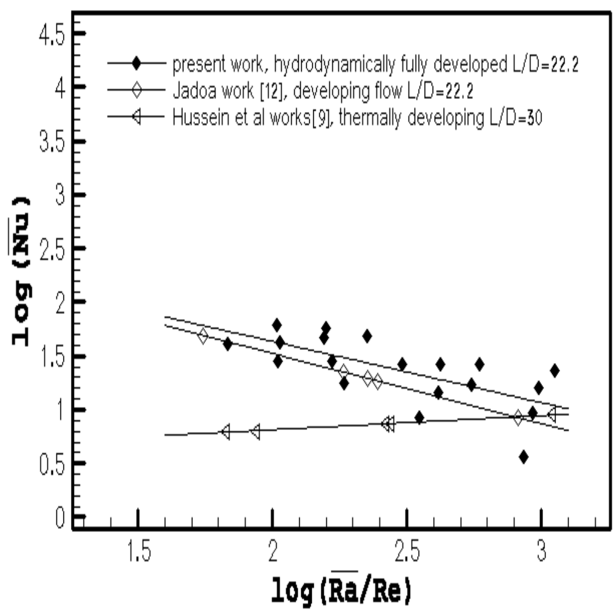


Figure.(12): Comparison of the average heat transfer result with available literature for α=60^0 (Inclined position)

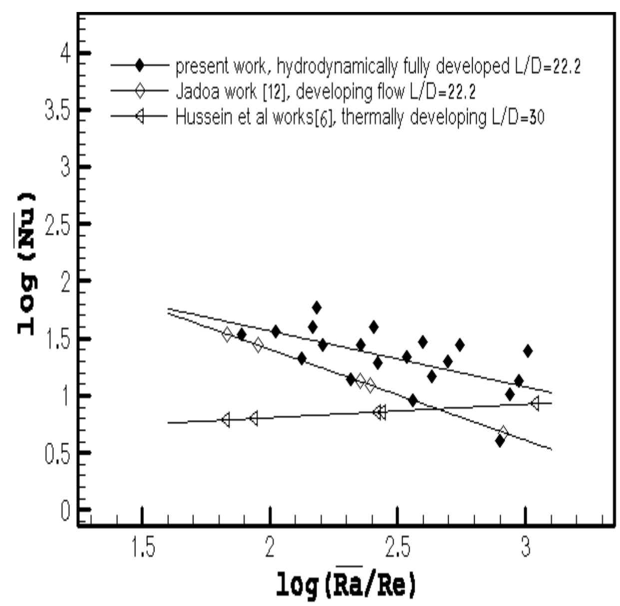


Figure.(13): Comparison of the average heat transfer results with available literature for α=90^0 (Vertical position)

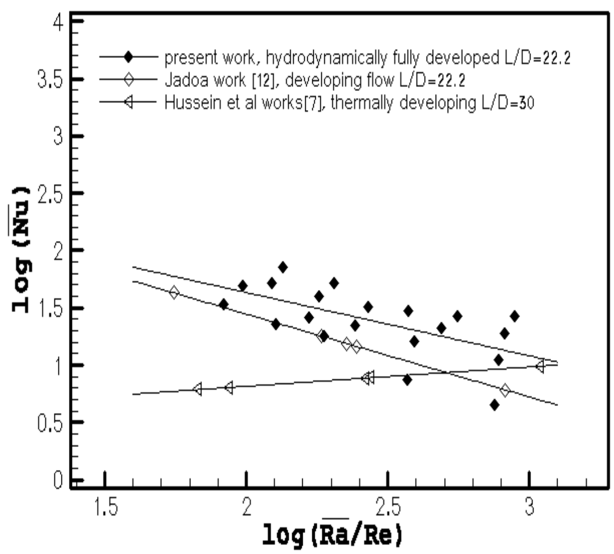
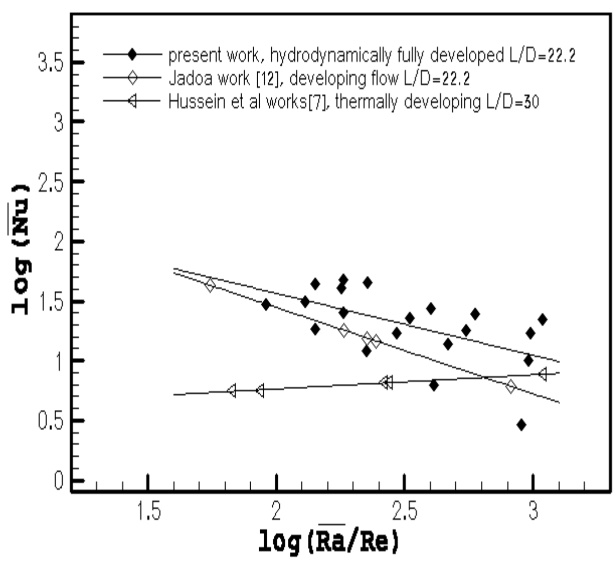
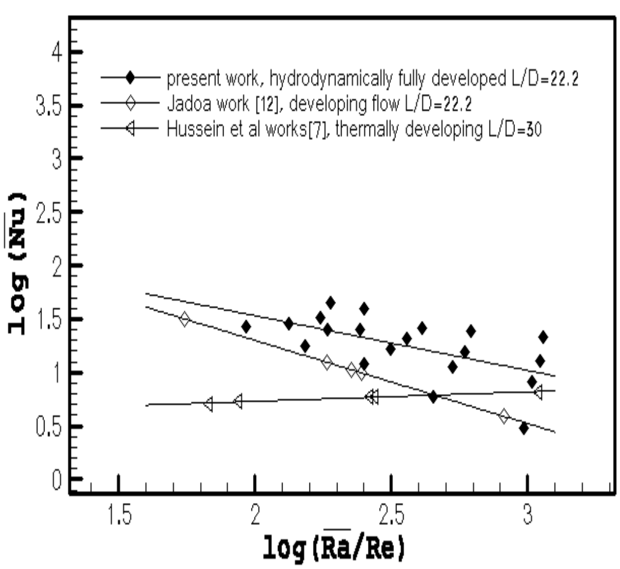


Figure.(14): Comparison of the average heat transfer result with available literature for α=-30^0 (Inclined position)



Figure(15): Comparison of the average heat transfer resul with available literature for α=-60^0 (Inclined position)



Figure(16): Comparison of the average heat transfer results with available literature for α=-90^0 (Vertical position)