European Journal of Scientific Research
ISSN 1450-216X / 1450-202X Vol. 97 No 1 February, 2013, pp.144-158
http://www.europeanjournalofscientificresearch.com

Mixed Convection Heat Transfer in Inclined Tubes with Constant Heat Flux

Ahmed Tawfeeq Ahmed Al-Sammarraie
Mechanical Engineering Department, College of Engineering
University of Tikrit, Tikrit, Iraq
E-mail: ahtawfeeq@gmail.com
Tel: +964 770 3769881

Raaid Rashad Jassem
Mechanical Engineering Department, College of Engineering
University of Tikrit, Tikrit, Iraq
E-mail: rrj5388@gmail.com
Tel: +964 781 0040167

Thamir K. Ibrahim
Mechanical Engineering Department, College of Engineering
University of Tikrit, Tikrit, Iraq
E-mail: thamirmathcad@yahoo.com
Tel: +964 771 1775455

Abstract

Mixed convection heat transfer in inclined tubes of circular cross section has been experimentally studied for assisting, thermally developing and thermally fully developed laminar-to-turbulent air flows, under constant wall heat flux boundary condition, at Reynolds numbers (Re<2300) for the laminar and (2300<Re<4000) for transition-to-turbulent air flows, and the heat flux is varied from 492 to 1442 W/m^2. The mixed convection regime has been bounded by the convenient selection of Reynolds number range and the heat flux range so that the obtained Richardson number (Ri) is varied approximately from 0.146 to 1.058. The experimental rig consists of three copper tubes as test section with 600 mm heated length and length to diameter ratio (L/D= 11.8, 15.75 and 31.5). This study has investigated the effect of the heat flux, diameters and inclination angle of the tube on the mixed convection heat transfer process. In this search, the local Nusselt numbers (Nu) with the dimensionless axial distance (Z) are presented. The results have clearly shown that an increase in the Nusselt number values as the heat flux increases and as the tube inclination angle moves from ($\theta = 60°$) to ($\theta = 30°$), vice versa the Nusselt number decrease with effect of increase the length to diameter ratio (L/D). For the range of Reynolds numbers used in experiments, the maximum Nusselt number has occurred at about 30° inclination relative to the horizon. Present experimental results have a good agreement with previous results obtained for similarly tubes inclination angles. Based on the experimental results, the average Nusselt number ($\overline{Nu}$) has been correlated in an empirical equation with the effect of the Rayleigh number, Reynolds number, length to diameter ratio and inclination angle. Good agreement can be seen between the experimental results and this equation.

Keywords: Mixed Convection Heat Transfer, Constant Heat Flux, Inclined Tube, Diameter.

1. Introduction

Convection heat transfer for thermally developing air flow in circular ducts is encountered in a wide variety of thermal engineering applications such as compact heat exchangers, cooling of electronic equipment, solar collectors, and thermal-energy conversion devices. The performance of these devices depends on the nature of heat dissipation and the operation modes of the system. In convective heat transfer problems, the flow that caused by external forces, such as pumps or fans is classified usually as forced convection, while the convection flows with effect fluid density variations due to the wall to fluid temperature difference under the influence of body forces are called free convection flows. The superposition of free convection on the forced convection heat transfer process gives rise to a new field of study called mixed convection. Therefore, the mixed convection situation extends from the extremes of the free convection regime on one hand, when the motion results from buoyancy alone, to the forced convection regime on the other hand, when external forces alone create the motion and buoyancy forces are insignificant (Mohammed and Salman, 2007).

The interaction of the free and forced convection currents can be extremely complex and difficult because it depends not-only on all the parameters determining both free and forced convection relative to one another. An understanding of the various fluid flow and heat exchange processes enables proficient design of these devices. Therefore, investigate on fluid flow and heat transfer through circular ducts requires significant attention. The nature of the flow, thermal boundary conditions and the diameter of the circular duct has a significant impact on the amount of heat energy transported by the working fluid in a circular duct. Air is a most common working fluid and is extensively preferred as a medium for cooling of electronic equipments, due to the behavior advantages of the air and its low cost. The application potential of the inclined of circular ducts underscores the importance of the study of mixed convection with surface radiation effects (Mohammed and Salman, 2007; Jackson et al, 1989; Yan and Li, 2001).

Comprehensive review of experimental studies has been presented on convection heat transfer in internal flows (Mohammed and Salman, 2007; Jackson et al, 1989). Experimental study was carried out to show the behavior of convection flow in a channel heated from one side wall (Gau et al, 2000; Yang et al, 2009). Convection laminar flow between parallel plates heated uniformly from below was investigated (Maughan and Incropera, 1987; Maughan and Incropera, 1990), and correlations were predicted for the fully developed Nusselt numbers and the locations of the inception of secondary flow. Many researchers (Mohammed and Salman, 2007; Chang and Lin, 1997; Dogan et al, 2005) performed extensive experimental studies on bottom heated horizontal ducts, horizontal circular cylinders, aspect ratio effects in a horizontal duct, hydrodynamically and thermally developed flow in a horizontal circular cylinder and a rectangular channel with separate heat sources.

Iqbal and Stachiewicz (1966) performed a theoretical analysis for fully developed upward laminar flow inside circular tube with constant wall heat flux and constant pressure gradient. The temperature, velocity, and Nusselt number were calculated by perturbation analysis. It was concluded that as the tube inclination varies from horizontal to vertical, Nusselt number increases up to a maximum value, which may occur before the vertical position is reached. Cheng and Hong (1973) applied a numerical solution using a combination of boundary vorticity method and line iterative relaxation method for upward fully developed laminar flow in tube subjected to thermal boundary conditions of axially uniform wall heat flux and peripherally uniform wall temperature at any axial position. The results showed that, in high Rayleigh number regime, the tube orientation effect had a significant effect on the results in the neighborhood of horizontal position. Sabbagh et al (1976) introduced experimental study for developing air flow in an inclined circular tube with uniform peripheral temperature and axial wall heat flux. The variation of the Nusselt number with tube inclination angles had been compared with a theoretical study done by (Iqbal and Stachiewicz, 1972), at low Rayleigh numbers and low Reynolds numbers.

Mare et al (2005) numerically solved, the elliptical coupled steady state three-dimensional governing partial differential equations for heated ascending laminar mixed convection in an inclined isothermal tube using a finite volume staggered grid approach, to determine the axial evolution of the hydrodynamic, thermal fields and investigate the presence of flow reversal. The effect of Grashof number on the axial evolution of the wall shear stress and Nusselt number was shown to be very important in the region of developing flow. The results had been calculated for one Reynolds number (Re = 100), a single fluid (air), and one tube inclination 45^o. Mohammed and Salman (2007) conducted an experimental study for the local and average heat transfer by mixed convection for hydrodynamically fully developed, thermally developing and thermally fully developed laminar air flow in an inclined circular cylinder. The results of surface temperature, the local and average Nusselt number distributions with the dimensionless axial distance were presented. For all entrance sections, the results showed an increase in the Nusselt number values as the heat flux increases and as the angle of cylinder inclination moves from $\theta = 60^o$ inclined cylinder to $\theta = 0^o$ horizontal cylinder. Mohammed and Salman (2009) investigated experimentally the effect of different inlet geometries on laminar air flow combined convection heat transfer inside a horizontal circular pipe. A wall boundary heating condition of a constant heat flux was imposed. It was observed that, the Nusselt number values for bell-mouth inlet geometry were higher than other inlet geometries due to the differences in the average temperatures and densities of the air. The average heat transfer results were correlated with an empirical correlation in terms of dependent parameters.

2. Object of Study

The purpose of the present study is to determine experimentally the effects of heat flux, tube inclination angles with the horizon and tubes diameters on the laminar-to-turbulent air flows heat transfer process, under mixed heat convection for assisting, thermally developing and thermally fully developed air flows situation in uniformly heated an inclined circular tube angle (30^o, 45^o and 60^o) and for different values of diameters (0.75, 1.5 and 2 inch).

3. Experimental Work

Published works on mixed convection heat transfer in inclined circular cross-section tubes dating to 1966, when Iqbal and Stachiewicz (1966) used perturbation analysis to calculate velocity and temperature fields for laminar, fully-developed, upward flow and uniform heat flux. Since then, the problem has been investigated experimentally by several research groups, but the published results cover a rather limited range of the independent parameters. The literature contains a comparable number of numerical studies, but once again, they are far from being exhaustive. The heat transfer by mixed convection is represented an important form of the convection heat transfer. An experimental test model accomplishes to measure the mixed convection in circular inclined ducts with different angles (30^o, 45^o and 60^o) heated with a constant heat flux.

The experimental rig is shown photographically in Figure (1) and schematically in Figure (2). It consists mainly of three copper tubes which were used in this study, they have internal diameters of (19.05, 38.1 and 50.8 mm) with wall thickness (2 mm). The length was (1000 mm) for each tube, but the active heated length (test section) only is (600 mm). The tubes were provided with air by a blower operates at (2500 rpm and AC 220 V).

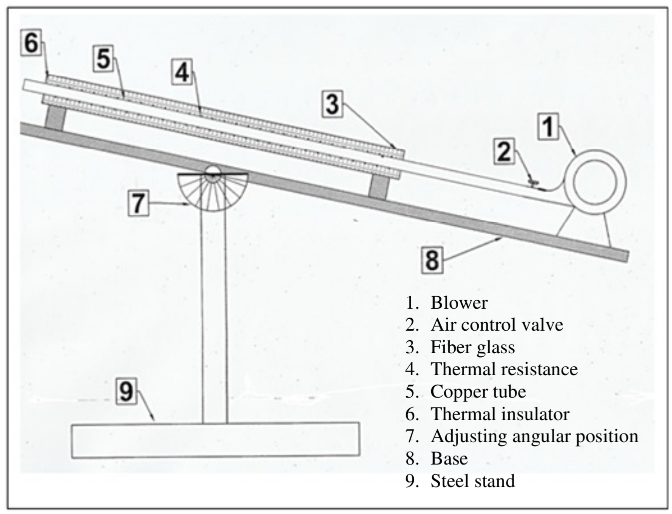
For each tube, surface temperatures were measured by (12) thermocouples (T type), which were positioned along the test section in the rate of one thermocouple for each location. Also, one thermocouple was used to measure the inlet air temperature which was located in the entrance of each tube, another one was located at the end of each tube to measure the outlet air temperature so that it was positioned in the same pervious way. Another thermocouple measures the laboratory ambient temperature.

In each tube, the test section was heated electrically using nickel-chrome wire with appropriate resistance for each tube diameter, it was electrically isolated by ceramic beads, and the wire was wounded uniformly along it as a coil in order to give uniform wall heat flux. The test section was thermally insulated with fiber glass of (100 mm) in thickness. The two ends of each tube were insulated electrically and thermally using two pieces made of Teflon, also they were used to reduce the thermal losses in the axial direction. The entrance sector was packed to be insuring to get hydrodynamically fully developed before the test section.

The heater was supplied with an alternative electrical power using voltage regulator (0 - 220 V) that supplied with a steady voltage through a stabilizer. The current pass through the heater and the voltage across both its ends were measured by a digital ammeter and an accurate voltmeter, respectively. Digital Anemometer was used to measure the velocity of entering air to the test section.

Figure 1: Photo of the experimental rig.

Figure 2: Schematic drawing for the test rig including the test sections.



The primary purpose of the test is to demonstrate the impact of heat flux, tube inclination angles and tubes diameters on mixed convection heat transfer through the test section. The procedure for that can be listed as follow:

1. In the beginning of any test, is determined the tube and its inclination angle with the horizon.
2. The air blower is turned on in order to supply the test section by the required amount of the air, which can be controlled using the control valve that shown in Figure (2). Waiting for few minutes to ensure that the hydrodynamic homogeneity state within test section is achieved.
3. Supply an elected electrical power to the electric heater by the voltage regulator and consequently obtained the amount of heat to be supplied to the test section.
4. The test rig is left for enough time (40- 45) minutes until accessing the steady state.
5. Then, the readings are taken for: the internal surface temperatures along the test section (T_{sz}), the inlet and outlet and ambient air temperature (T_i, T_o, T_a), the amount of voltage (V) and the current (I), and outlet air velocity (U) which it fixed at a constant value in the present work.
6. After that, the electrical power supplied to the heaters is altered with another amount through the voltage regulator and the process is repeated again.

4. Calculation Procedures

The results obtained were reduced from the present experimental work -for each tube diameter- in terms of local Nusselt number (Nu) with the dimensionless axial distance (Z=z/L), as a function of the tube inclination angle (θ) at constant wall heat flux boundary condition. Then, the total input power supplied to the test section can be calculated using the following equation:

$$Q_t = V \times I \tag{1}$$

The convection heat transferred from the test section surface:

$$Q_{conv.} = Q_t - Q_{cond.} \tag{2}$$

where $Q_{cond.}$ are the total conduction heat losses (lagging and ends losses).

The convection heat flux can be represented by:

$$q_{conv.} = \frac{Q_{conv.}}{A_s} \tag{3}$$

where A_s is the surface area ($A_s = \pi \times D \times L$), D is the test section diameter, and L it is length.

The local heat transfer coefficient (h_z) was calculated by the following equation:

$$h_z = \frac{q_{conv.}}{T_{sz} - T_{bz}} \tag{4}$$

where T_{sz} is the local surface temperature, and T_{bz} is the local bulk air temperature which can be evaluated as reported by (Peyghambarzadeh, 2011):

$$T_{bz} = T_i + \frac{q_{conv.}\pi D}{mC_P} z \tag{5}$$

where m is the air mass flow rate ($m = \rho U(\pi D^2/4)$).

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

$$Nu = \frac{h_z D}{k_{fz}} \tag{6}$$

where k_{fz} is the local film air thermal conductivity, which can be evaluated at the local film air temperature. Then, the local film air temperature (T_{fz}) is presented by:

$$T_{fz} = \frac{T_{sz} + T_{bz}}{2} \tag{7}$$

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

$$T_s = \frac{1}{L}\int_{z=0}^{z=L} T_{sz}dz \text{ , } T_b = \frac{1}{L}\int_{z=0}^{z=L} T_{bz}dz = \frac{T_i + T_o}{2} \text{, and } T_f = \frac{T_s + T_b}{2} \tag{8}$$

then:

$$\overline{Nu} = \frac{hD}{k} = \frac{q_{conv.} \times D}{k(T_s - T_b)} \tag{9}$$

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

Reynolds number: $$\mathrm{Re} = \frac{\rho U D}{\mu} \tag{10}$$

Grashof number: $$Gr = \frac{g\beta(T_s - T_b)D^3}{\upsilon^2} \tag{11}$$

where g is the acceleration of gravity (9.81 m/s^2) and $\beta = \dfrac{1}{(T_f + 273)}$

Rayleigh number: $$Ra = Gr \times \mathrm{Pr} \tag{12}$$

Richardson number: $Ri = \frac{Gr}{\mathrm{Re}^2}$ (13)

All the air physical properties ($C_P, \rho, k, \mu, \upsilon,$ and Pr) were evaluated at the film air temperature as reported by (Incropera and DeWitt, 2003).

Kline and McClintock method (Holman, 2012) was used for the experimental error analysis of heat transfer coefficient (Nusselt number), Reynolds number, Grashof number, Rayleigh number and Richardson number. The error was limited between (2%) to (11%) for all experimental data.

5. Results and Discussion

Mixed convection heat transfer from inclined tubes of circular cross section under constant heat flux condition has been studied. The effects of three different parameters: heat flux, inclination angles of the tube and tube diameter on the mixed convection heat transfer from the internal surface of the tube were investigated. To investigate steady state mixed convection heat transfer represented by Richardson number (Ri) bounded by the range of ($0.1 < Ri < 10$) through inclined tubes at Reynolds number (Re<2300) for the laminar and (2300<Re<4000) for transition-to-turbulent air flows under thermally developing and thermally fully developed air flows, an experimental rig was constructed.

The obtained Richardson numbers (Ri) from this study is varied approximately from 0.146 to 1.058. This range means that the mixed convection regime has been bounded by the suitable selection of the Reynolds number range (the air flow rate) and the Grashof number range (the input power) in the current study.

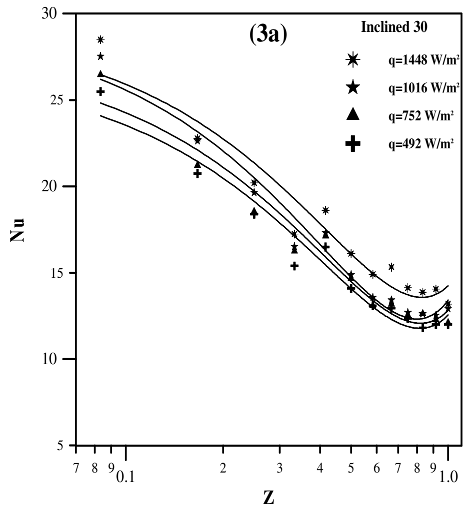
In the present work, Figures from (3) to (11) were represented the results in terms of local Nusselt number (Nu) with the dimensionless axial distance (Z):

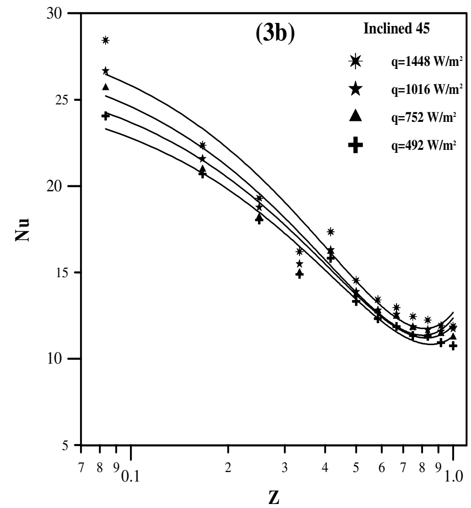
a. Figures (3, 4, 5) highlights the influence of the heat flux on heat transfer from certain tube diameter - 0.75 in (L/D = 31.5), 1.5 in (L/D = 15.75), or 2 in (L/D = 11.8) - at various inclination angles ($\theta = 30^o, 45^o, 60^o$).
b. Figures (6, 7, 8) highlights the influence of the inclination angle on heat transfer from certain tube diameter - 0.75 in (L/D = 31.5), 1.5 in (L/D = 15.75), or 2 in (L/D = 11.8) - at various heat fluxes (492, 752, 1016, 1448 W/m^2).
c. Figures (9, 10, 11) highlights the influence of the tube diameter on heat transfer from certain inclination angles ($\theta = 30^o, 45^o, or 60^o$) at various heat fluxes (492, 752, 1016, 1448 W/m^2).

For all these Figures, it is obvious that the local Nusselt number values decreased with increase of the dimensionless axial distance of the tube. This reveals that the local Nusselt number near the inlet of the tube heated region (test section) are very high values because the thickness of the thermal boundary layer is zero, and it decreases continuously due to the thermal boundary layer develops and then near the exit of the tube heated region the local Nusselt number values slightly increases due to the laminarization effect in the near wall region (buoyancy effect) and tube end losses.

Regarding Figures (3, 4 and 5), it is noticed that the local Nusselt number values increased with increase of the heat flux for all length to diameter ratios (tested tubes diameters) at various inclination angles. This is due to the fact that, the increase in the heat flux leads to increase the energy added to the fluid flowing through the test section, and it may be attributed to the secondary flow superimposed on the forced flow which its effect increase as the heat flux increases leading to a higher heat transfer coefficient, also due to the free convection currents domination on the heat transfer process. This result was consistent with the results that reported by (Mohammed and Salman, 2007; Mohammed and Salman, 2009).

Figure 3: The relation between local Nusselt number and the dimensionless axial distance for heat fluxes of the ratio (L/D = 31.5) at various inclination angles (a, b, c).





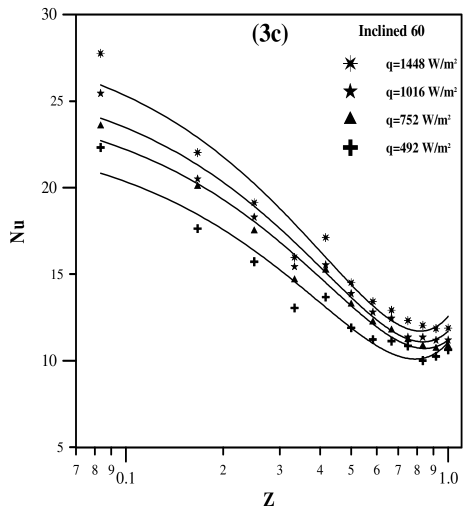


Figure 4: The relation between local Nusselt number and the dimensionless axial distance for heat fluxes of the ratio (L/D = 15.75) at various inclination angles (a, b, c).

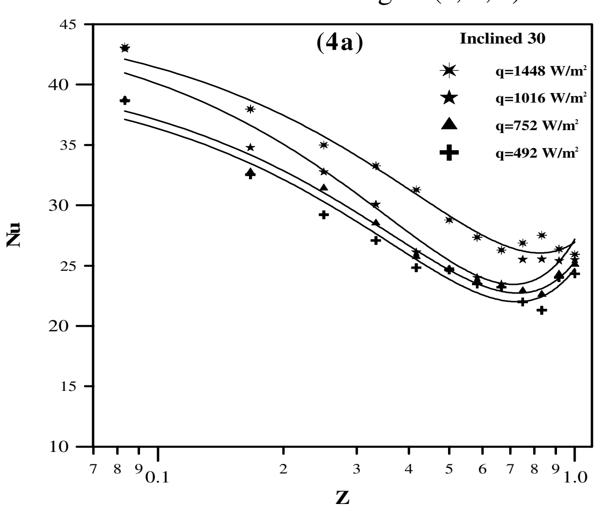


Figure 5: The relation between local Nusselt number and the dimensionless axial distance for heat fluxes of the ratio (L/D = 11.8) at various inclination angles (a, b, c).

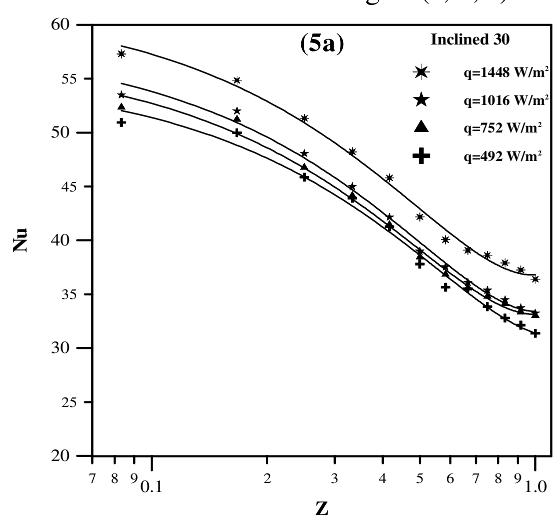
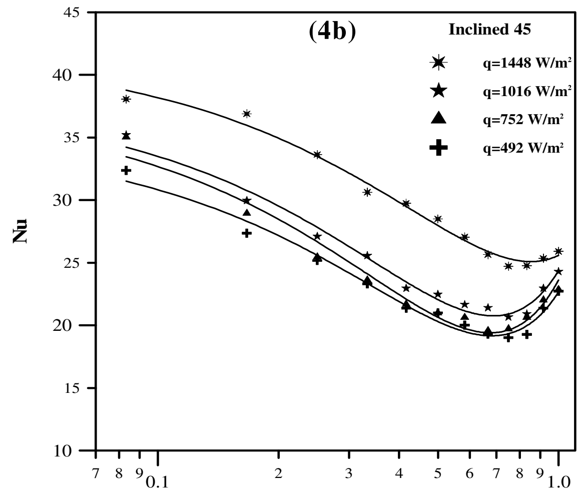


Figure 4: The relation between local Nusselt number and the dimensionless axial distance for heat fluxes of the ratio (L/D = 15.75) at various inclination angles (a, b, c). - continued



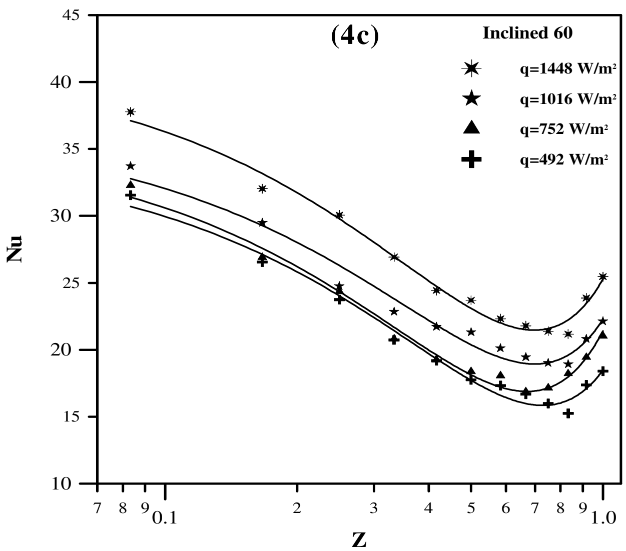
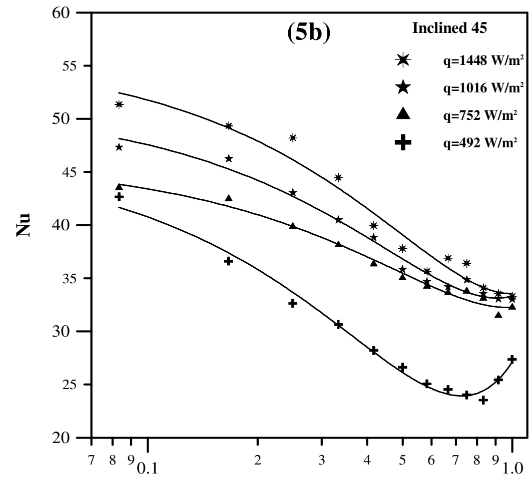
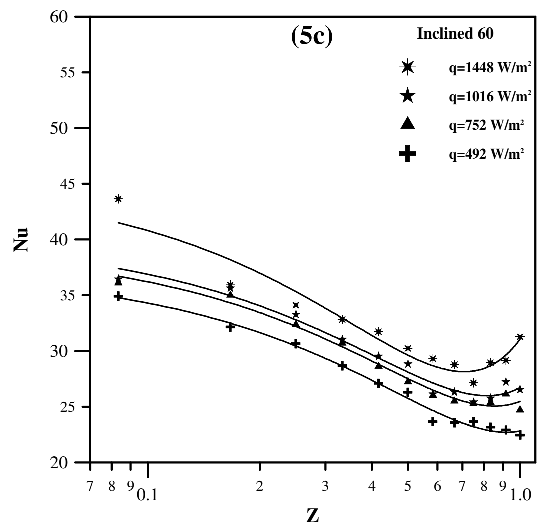


Figure 5: The relation between local Nusselt number and the dimensionless axial distance for heat fluxes of the ratio (L/D = 11.8) at various inclination angles (a, b, c). - continued





As for Figures (6, 7 and 8), it is clear that the local Nusselt number values increased with decrease of the inclination angle values for all length to diameter ratios (tested tubes diameters) at various heat fluxes. In more detail, it is noticed from the Figures the higher local Nusselt number has occurred with tube inclination angle (30^o) and its lower value has appeared with tube inclination angle (60^o). It was because that the increase of inclination angle led to increase the impact of buoyancy force on the fluid flow through the mixed convection heat transfer condition ($0.1 < Ri < 10$). This remarkable buoyancy effect causes an obstruction fluid flow in the tube and reducing the fluid discharge slightly. In addition, growth the backward flow cause increased the time that necessary to absorb the heat from the tube by the fluid flowing inside it. In other words, the local Nusselt number values increased as the inclination angle moves toward to the horizon due to the increase of the secondary flow, which enhances the heat transfer process.

Figure 6: The relation between local Nusselt number and the dimensionless axial distance for inclination angles of the ratio (L/D = 31.5) at various heat fluxes (a, b, c, d).

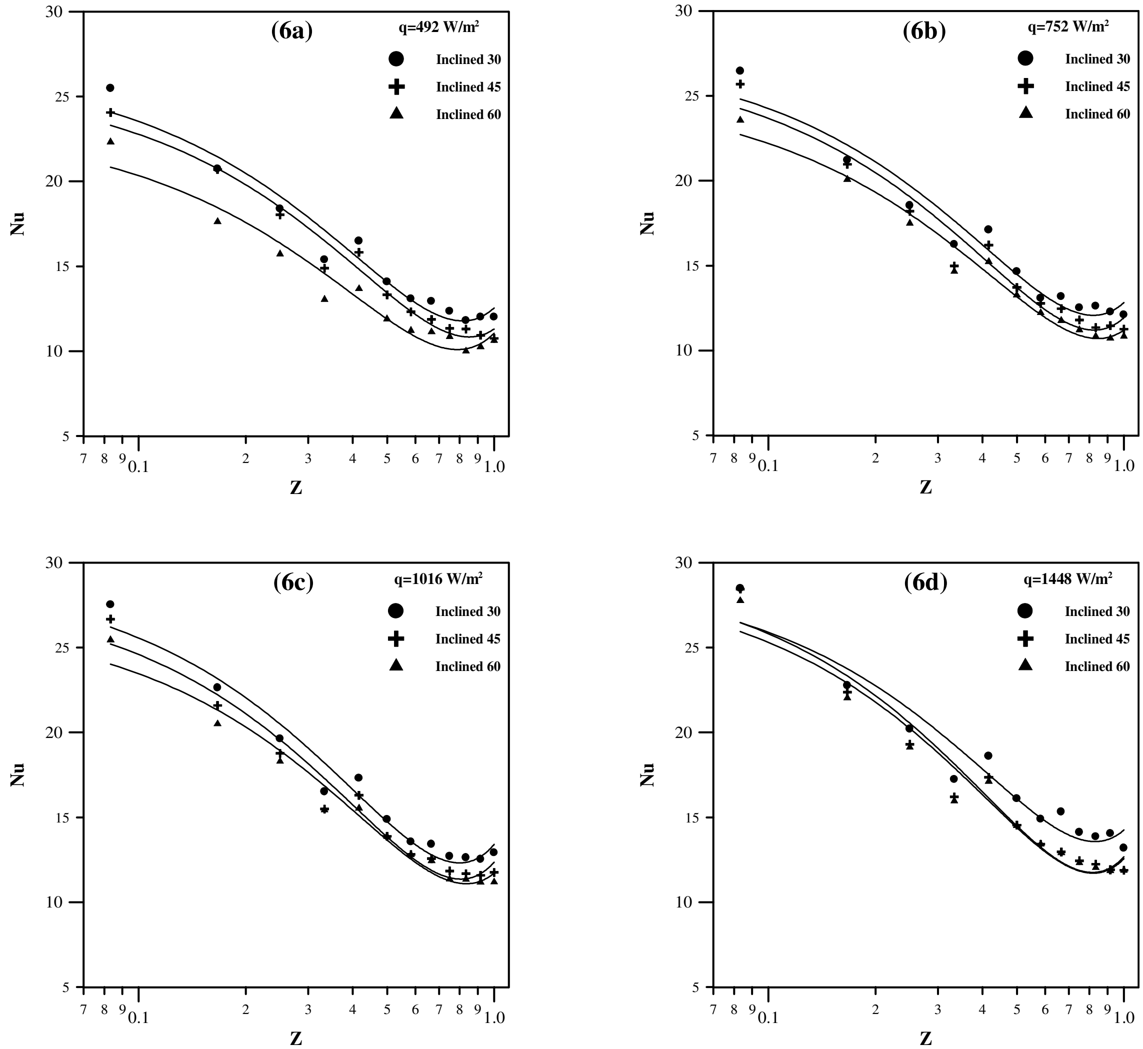


Figure 7: The relation between local Nusselt number and the dimensionless axial distance for inclination angles of the ratio (L/D = 15.75) at various heat fluxes (a, b, c, d).

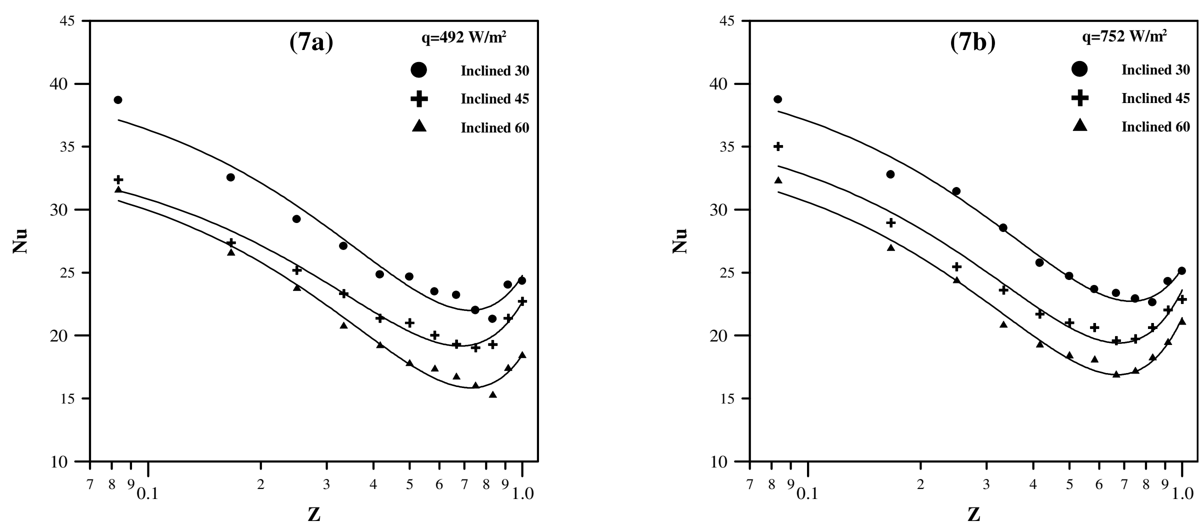


Figure 7: The relation between local Nusselt number and the dimensionless axial distance for inclination angles of the ratio (L/D = 15.75) at various heat fluxes (a, b, c, d). - continued

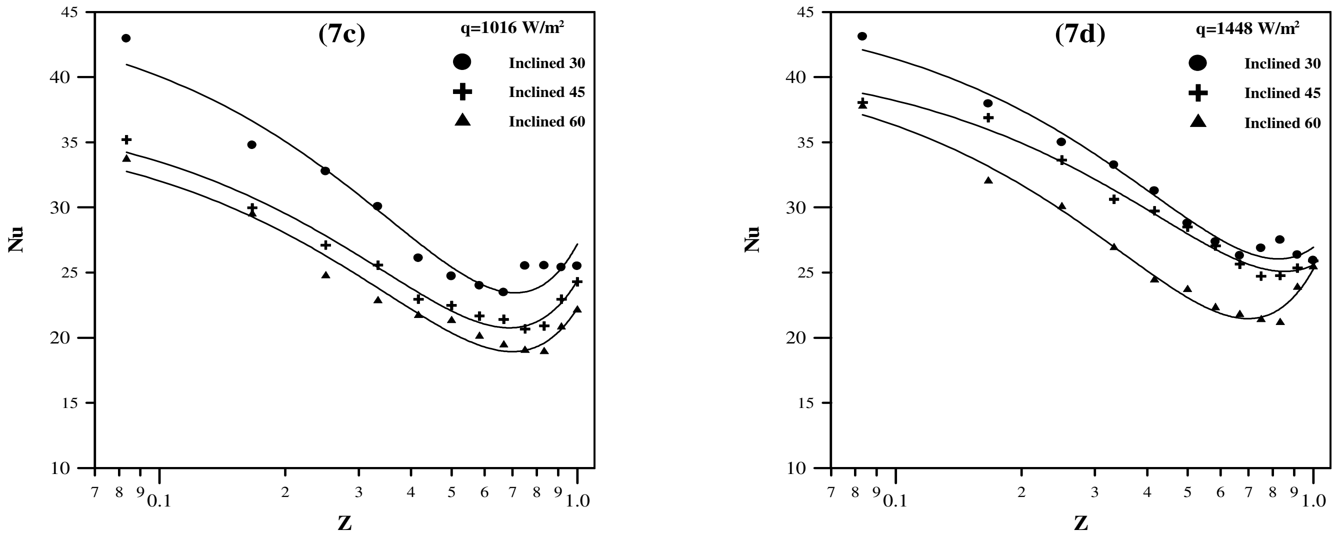
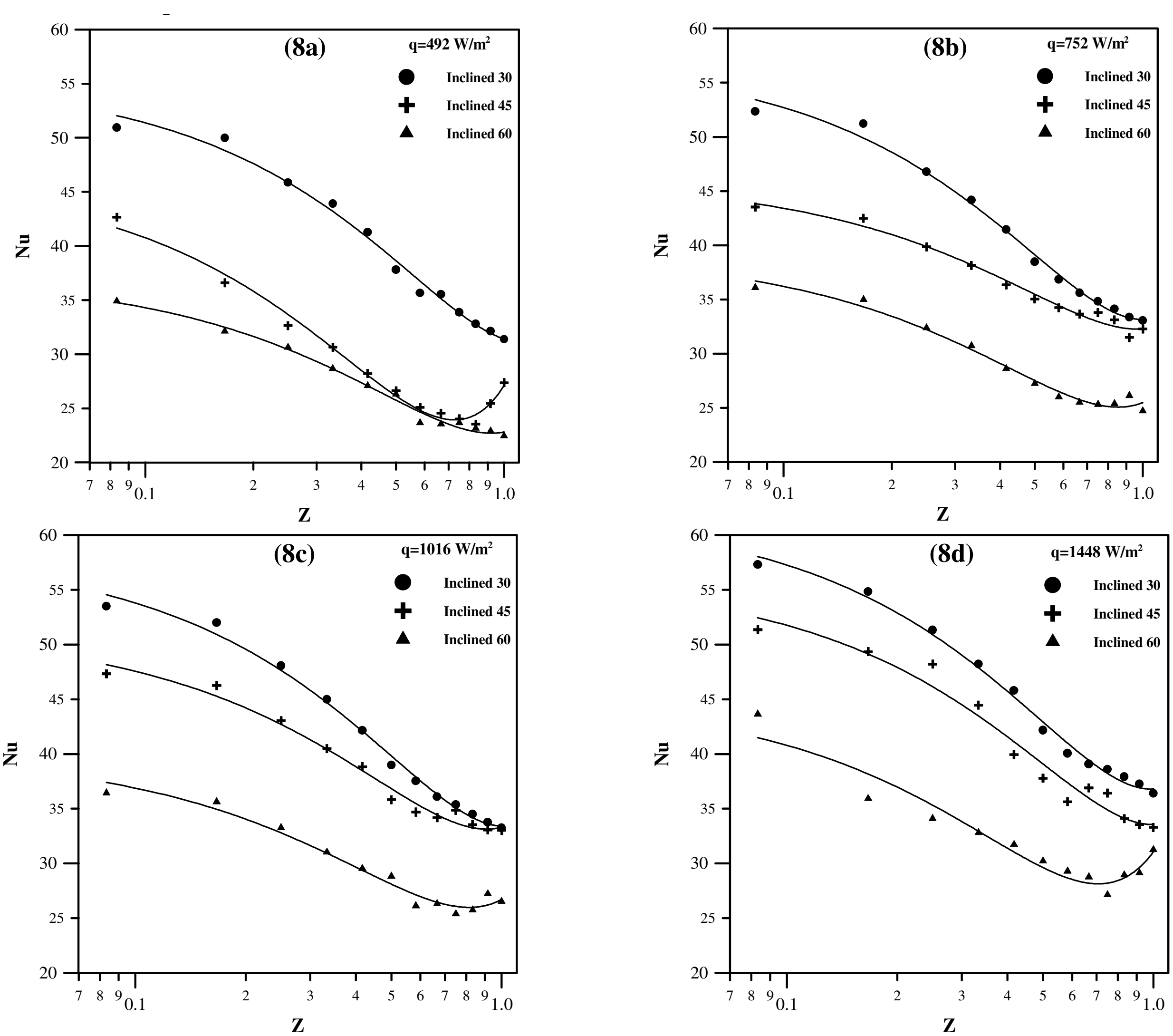


Figure 8: The relation between local Nusselt number and the dimensionless axial distance for inclination angles of the ratio (L/D = 11.8) at various heat fluxes (a, b, c, d).



From Figures (9, 10 and 11), it is evident that the local Nusselt number values increased with increase of the tube diameter (with decrease the length to diameter ratio) for all inclination angles at various heat fluxes, i.e., the higher local Nusselt number value is observed at tube diameter (2 in (L/D=11.8)) while the lower value is noticed at tube diameter (0.75 in (L/D=31.5)). It is because that the increased of the tube diameter -with fixed values of tube length, heat flux and inclination angle of the tube- cause increased the heat transfer surface area between the tube inner surface and the fluid flowing through it. So this increase of the heat exchange surface gave the flowing fluid a good opportunity to contact with tube inner surface, and then increased the heat transfer process. On the other hand, this increase for the tube diameter -with fixed all other values- leads to increase of Reynolds numbers which leading to the transformation in the flow from the laminar at (L/D=31.5), transition-to-turbulent at (L/D = 15.75) and turbulent at (L/D=11.8), respectively. The transformation in the shape of flow from laminar to turbulent is a significant effect in increasing the rate of heat transfer from the heat exchange surface. So this influence was clear on the results of the current study.

Figure 9: The relation between local Nusselt number and the dimensionless axial distance for all tubes diameters at inclination angle ($\theta = 30^o$) at various heat fluxes (a, b, c, d).

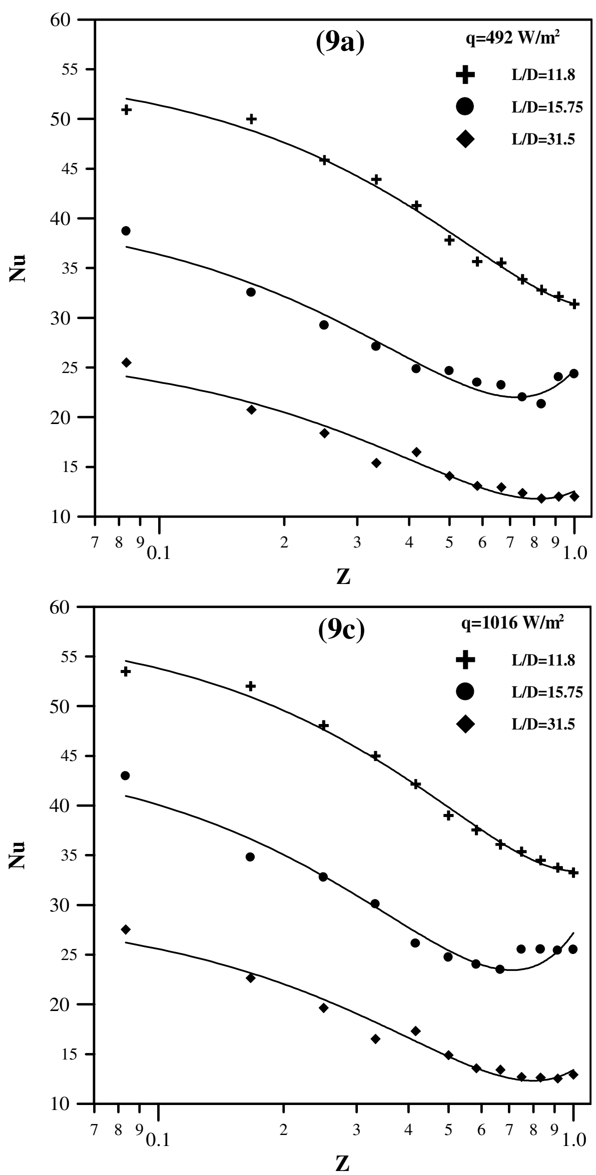


Figure 10: The relation between local Nusselt number and the dimensionless axial distance for all tubes diameters at inclination angle ($\theta = 45^o$) at various heat fluxes (a, b, c, d).

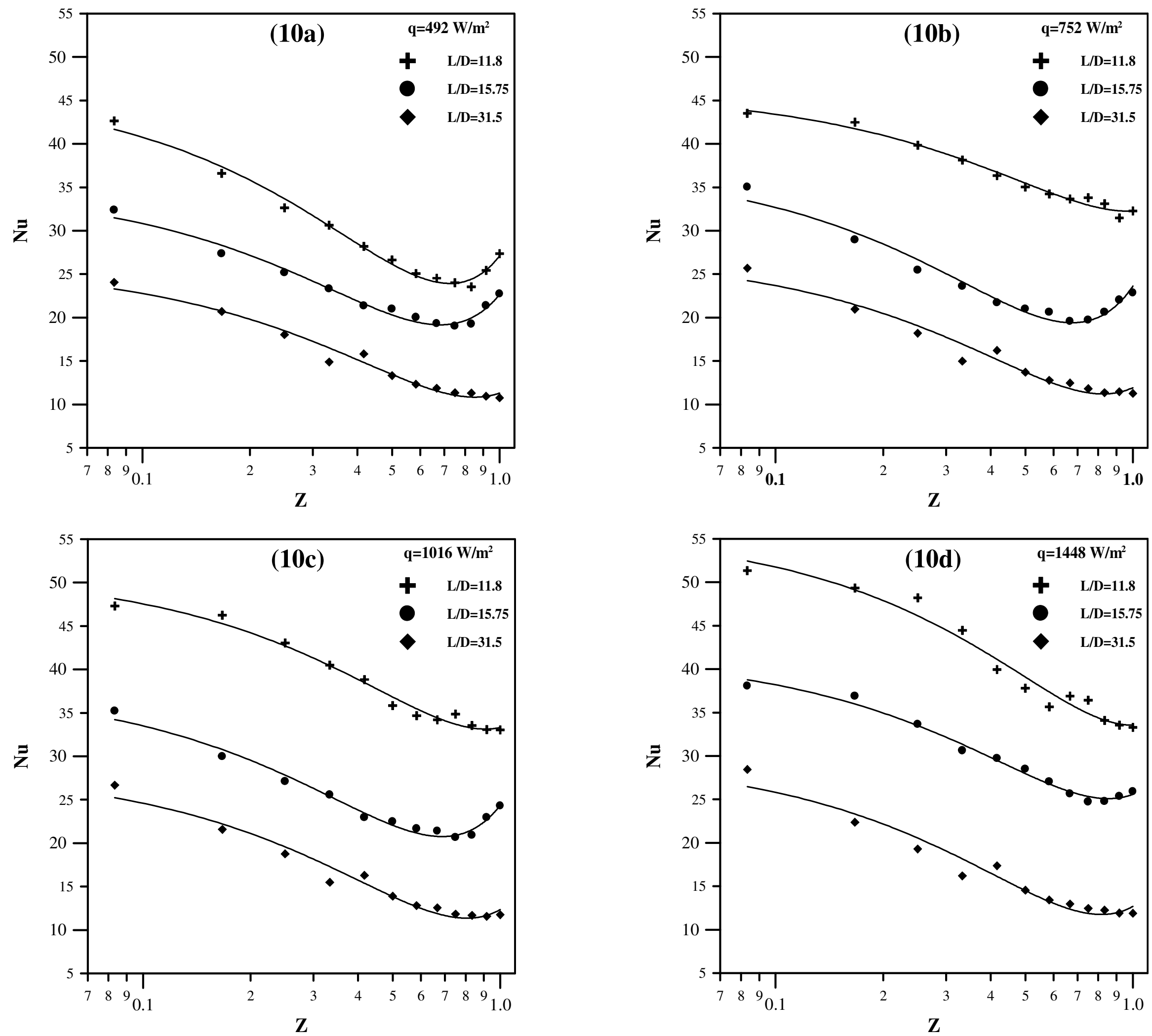


Figure 11: The relation between local Nusselt number and the dimensionless axial distance for all tubes diameters at inclination angle ($\theta = 60^o$) at various heat fluxes (a, b, c, d).

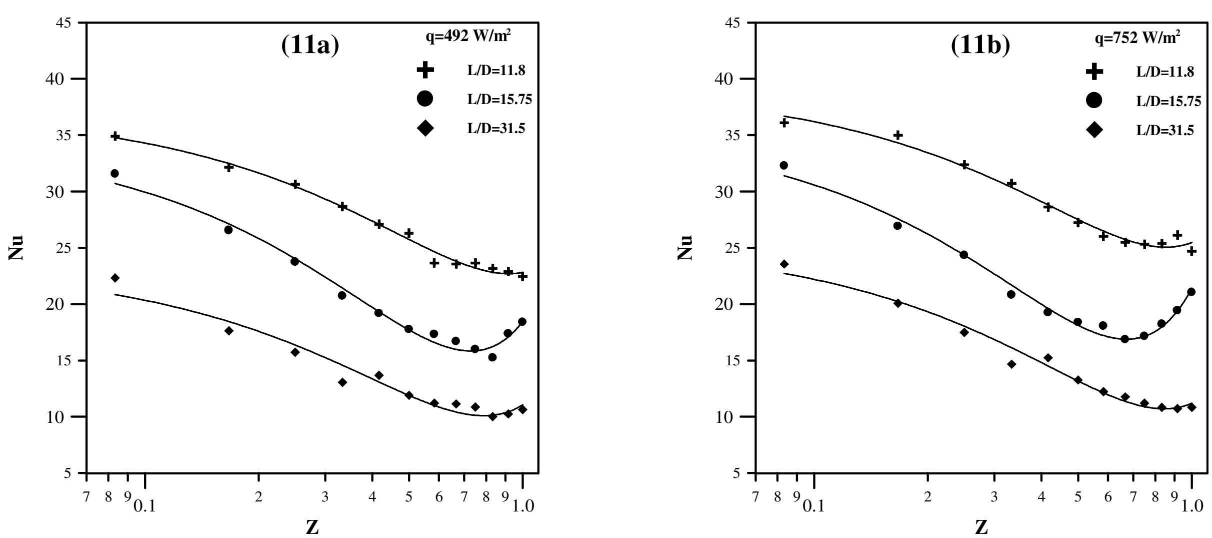
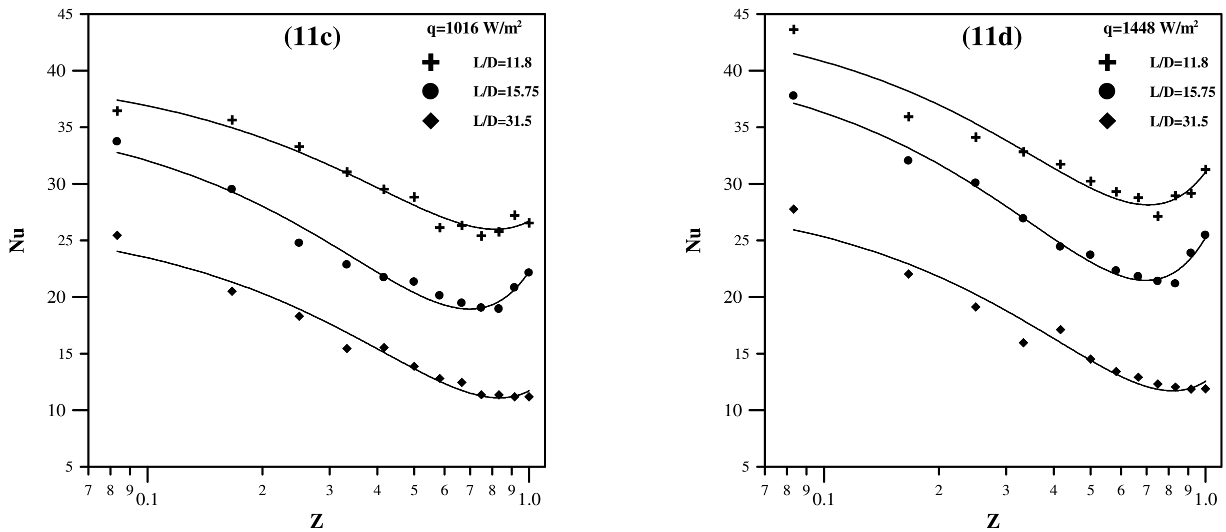


Figure 11: The relation between local Nusselt number and the dimensionless axial distance for all tubes diameters at inclination angle ($\theta = 60^o$) at various heat fluxes (a, b, c, d). - continued



An empirical formula was extracted for the experimental data of mixed convection heat transfer with the effective parameters on this process. The least squares fitting method was used for this purpose. The average heat transfer from any tested tube was represented by average Nusselt number ($\overline{Nu}$) as a function of the Rayleigh number to Reynolds number ratio (Ra/Re), length to diameter ratio (L/D) and the tube inclination angle with the horizon (θ). The correlating equation can be expressed as:

$$\overline{Nu} = a\left(Ra/\mathrm{Re}\right)^b \left(L/D\right)^c (1+\cos\theta)^d \tag{14}$$

Where a, b, c, and d represented the experimental constants, it can be calculated by applied the least squares fitting method on the experimental results. The experimental constants were calculated from this method, and it was as follows:

a = 19.59, b = 0.174, c = -0.567, d = 1.286

Therefore, the general formula of the equation that used to calculate the mixed convection heat transfer for all tested tubes is expressed as:

$$\overline{Nu} = 19.59\left(Ra/\mathrm{Re}\right)^{0.174} \left(L/D\right)^{-0.567} (1+\cos\theta)^{1.286}, \text{ at } \begin{Bmatrix} 30^o \le \theta \le 60^o \\ 11.8 \le L/D \le 31.5 \end{Bmatrix} \tag{15}$$

Figure (12) shows the relationship between heat transfer results and their own correlating equation for all the tubes diameters at all inclination angles. It is obvious that, there is a good agreement between the experimental data and correlating equation.

The graphical presentation of predicted results versus experimental data of the average values of Nusselt number is presents in Figure (13). It is clear that, for thirty six reading points that (89%) of these points are located within the deviation range of ($\pm 15\%$) from the correlating equation and this shows how the compatibility was good between this formula and the experimental results.

Figure 12: The relation between heat transfer results and their own correlating equation for all the tubes diameters at all inclination angles.

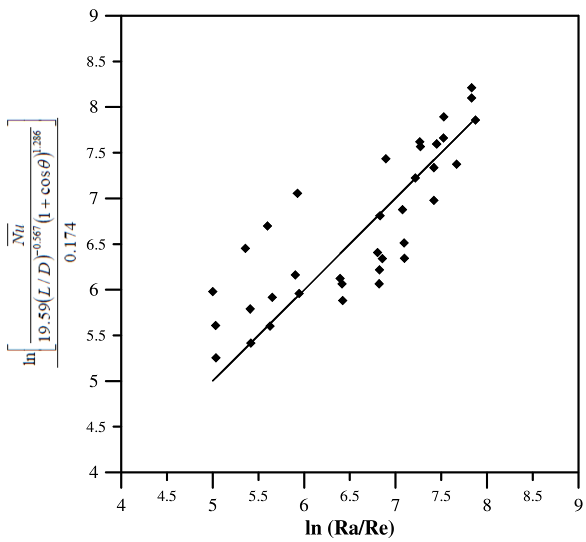
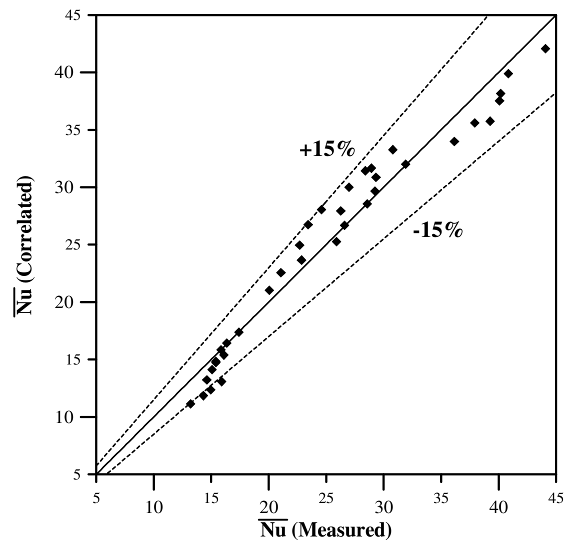


Figure 13: The comparison between measured Nusselt number and calculated Nusselt number from correlating equation.



6. Summary and Concluding Remarks

Mixed convection heat transfer in uniformly heated inclined circular tubes for assisting air flow with different diameters and same sections length, under constant wall heat flux, has been experimentally studied. The conclusions of results were summarized as follows:

1. The mixed convection regime has been bounded by the suitable selection of Reynolds number range and the heat flux range. The obtained Richardson numbers is varied approximately from 0.146 to 1.058.
2. The local Nusselt number values decreased along the tube test section as a result of thermally developing process along it.
3. For the same inclination angle and length to diameter ratio, the local Nusselt number values increased as the heat flux increases.
4. For the same heat flux and length to diameter ratio, the local Nusselt number values increased as the tube inclination angle moves from =60° toward =30°. The maximum and minimum amount of heat transfer occurs at about =30° and =60° from the horizontal axis, respectively.
5. For the same heat flux and tube inclination angle, the local Nusselt number values decreased as the length to diameter ratio increases.
6. A correlating equation for assisting flow, Eq. (15), has been derived to evaluate the average Nusselt number in terms of Rayleigh number, Reynolds number, length to diameter ratio and the tube inclination angle, with overall accuracy in order of (±15%).
7. Mixed convection heat transfer results have been compared with available literature and showed satisfactory agreement.

Acknowledgements

The authors would like to thank University of Tikrit for providing laboratory facilities and financial support.

References

[1] Mohammed, H. A., and Salman, Y. K., 2007. “Free and Forced Convection Heat Transfer in the Thermal Entry Region for Laminar Flow inside a Circular Cylinder Horizontally Oriented”, *Energy Conversion and Management* 48, pp. 2185-2195.

[2] Jackson, J. D., Cotton, M. A., and Axcell, B. P., 1989. “Studies of Mixed Convection in Vertical Tubes”, *International Journal of Heat and Fluid Flow* 10, pp. 1-15.

[3] Yan, W. M., and Li, H. Y., 2001. “Radiation Effects on Mixed Convection Heat Transfer in a Vertical Square Ducts”, *International Journal of Heat and Mass Transfer* 44, pp. 1401-1410.

[4] Gau, C., Jeng, Y. C., and Liu, C. G., 2000. “An Experimental Study on Mixed Convection in a Horizontal Rectangular Channel Heated from a Side”, *Transactions of the ASME Journal of Heat Transfer* 122, pp. 701-707.

[5] Yang, C. S., Liu, C. G., and Gau, C., 2009. “Study of Channel Divergence on the Flow and Heat Transfer in Horizontal Ducts Heated from a Side”, *International Journal of Thermal Sciences* 48, pp. 105-113.

[6] Maughan, J. R., and Incropera, F. P., 1987. “Experiments on Mixed Convection Heat Transfer for Air Flow in a Horizontal and Inclined Channel”, *International Journal of Heat and Mass Transfer* 30, pp. 1307-1318.

[7] Maughan, J. R., and Incropera, F. P., 1990. “Regions of Heat Transfer Enhancement for Laminar Mixed Convection in a Parallel Plate Channel”, *International Journal of Heat and Mass Transfer* 33(3), pp. 555-570.

[8] Chang, M. Y., and Lin, T. F., 1997. “Experimental Study of Aspect Ratio Effects on Longitudinal Vortex Flow in Mixed Convection of Air in a Horizontal Rectangular Duct”, *International Journal of Heat and Mass Transfer* 41, pp. 719-733.

[9] Dogan, A., Sivrioglu, M., and Baskaya, S., 2005. “Experimental Investigation of Mixed Convection Heat Transfer in a Rectangular Channel with Discrete Heat Sources at the Top and at the Bottom”, *International Communications in Heat and Mass Transfer* 32, pp. 1244-1252.

[10] Iqbal, M., and Stachiewicz, J. W., 1966. “Influence of Tube Orientation on Combined Free and Forced Laminar Convection Heat Transfer”, *Journal of Heat Transfer ASME Transactions* 88, pp. 109-116.

[11] Cheng, K. C., and Hong, S. W., 1973. “Combined Free and Forced Laminar Convection in Inclined Tube”, *Applied Science Research* 27 (1), pp. 19-38.

[12] Sabbagh, J. A., Aziz, A., El-Ariny, A. S., and Hamad, G., 1976. “Combined Free and Forced Convection Heat Transfer in Inclined Circular Tubes”, *Journal of Heat Transfer ASME Transactions* 98, pp. 322-324.

[13] Mare, T., Galanis, N., Pretot, S., and Miriel, J., 2005. “Mixed Convection with Flow Reversal in the Entrance Region of Inclined Tubes”, *International Journal of Numerical Methods for Heat and Fluid Flow* 15 (7), pp. 740-756.

[14] Mohammed, H. A., and Salman, Y. K., 2007. “Combined Convection Heat Transfer for Thermally Developing Aiding Flow in an Inclined Circular Cylinder with Constant Heat Flux”, *Applied Thermal Engineering* 27, pp. 1236-1247.

[15] Mohammed, H. A., and Salman, Y. K., 2009. “The Effect of Different Inlet Geometries on Laminar Flow Combined Convection Heat Transfer Inside a Horizontal Circular Pipe”, *Applied Thermal Engineering* 29, pp. 581-590.

[16] Peyghambarzadeh, S. M., 2011. “Forced Convection Heat Transfer in the Entrance Region of Horizontal Tube under Constant Heat Flux”, *World Applied Sciences Journal* 15 (3), pp. 331-338.

[17] Incropera, F. P., and DeWitt, D. P., 2003. “Fundamentals of Heat and Mass Transfer”, *5th. Edith, John Wiley & Sons Inc*.

[18] Holman, J. P., 2012. “Experimental Methods for Engineers”, *8th. Edition, McGraw- Hill Book Company, New York, USA*.